TP 7556E

COMBUSTION CHARACTERISTICS OF PROPANE DIRECTLY INJECTED INTO DIESEL ENGINES

Final Report

by

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Revision A

June 1986

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Un sommaire en francais de ce document est inclus avant la Table des matieres.

Transport Canada

Transports Canada

Development Développement

REPORT DOCUMENTATION FORM

1 Transport Canada Report No	2 TDC Project No	3 Recipient's Catalogue No
TP7556E	6060	

4 Title and Subtitle

Combustion Characteristics of Propane Directly Injected into Diesel Engines

5 Report Date
June, 1986

6 Performing Organization Report No.

7 Author(s)

U. Oester, J.S. Wallace, L. Segal and J.F. Keffer

8 Transport Canada File No.

9 Performing Organization Name and Address

Engine Research and Development Laboratory
Dept. of Mechanical Engineering
University of Toronto, 5 King's College Road
Toronto, Ontario M5S 1A4

10 DSS File No.
14SD.T8200-3-3533

11 DSS or Transport Canada Contract No
OSD83-00106

12 Sponsoring Agency Name and Address

Transportation Development Centre of Transport Canada
Suite 601, West Tower
200 Dorchester Blvd. West
Montreal, Quebec H2Z 1X4

13 Type of Report and Period Covered
FINAL

14 Sponsoring Agency Code

15 Supplementary Notes

Sponsored by the Energy R&D Panel

16 TDC Project Officer
J.H. Morgan

17 Abstract

A single-cylinder prechamber diesel engine was used to investigate propane fueling of diesel engines. Liquid propane was pumped to a high pressure and injected directly into the engine's prechamber at approximately the same time as diesel fuel would be injected in the original diesel engine. A continuously operating glow plug was used to provide ignition since propane has a very low cetane number. Tests were carried out at three compression ratios, 19:1, 22:1 and 25:1 and at three engine speeds, 1200 , 1500 and 1800 rpm.

For the same level of smoke opacity as the original diesel engine, the liquid propane injection engine produced about 15% more power. The relative position of the glow plug and fuel spray was critical to obtaining reliable ignition and short ignition delays. The final arrangement tested provided excellent results throughout the engine's operating range at 25:1 CR. However, at the two lower compression ratios, ignition was irregular at loads below about two-thirds of full load. At 25:1 compression ratio, the full load indicated specific fuel consumptions were 11 MJ/kW×h, 9 MJ/kW×h and 8 MJ/kW×h for diesel, liquid propane injection and propane fumigation (carbureted propane, pilot diesel ignition) respectively.

While additional development work is required, this project has demonstrated the feasibility and promise of the liquid propane injection engine with glow plug.

18 Key Words	19 Distribution Statement
Propane Diesel Engines Injection	Limited number of copies available from Transportation Development Centre on request

20 Security Classification (of this report)	21 Security Classification (of this page)	22 No. of Pages	23 Price
Unclassified	Unclassified		

TDC/CDT 79-005
Rev 86

Transport Canada
Développement Development

FORMULE DOCUMENTAIRE DU RAPPORT

1 N° du rapport Transports Canada	2 N° de l'étude — CDT	3 N° de catalogue du destinataire
TP7556E	6060	

4. Titre et sous-titre	5 Date du rapport
Combustion Characteristics of Propane Directly Injected into Diesel Engines	Juin, 1986
	6. N° du rapport de l'organisme exécutant

7. Auteur(s)	8 N° de dossier Transports Canada
U. Oester, J.S. Wallace, L. Segal and J.F. Keffer	

9 Nom et adresse de l'organisme exécutant	10 N° de dossier — ASC
Engine Research and Development Laboratory Dept. of Mechanical Engineering University of Toronto, 5 King's College Road Toronto, Ontario M5S 1A4	14SD. T°200-3-3533
	11. N° de contrat — ASC ou Transports Canada OSD83-00106

12. Nom et adresse de l'organisme parrain	13. Genre de rapport et période visée
Centre de Developpement de Transports, Transport Canada Suite 601, Tour Ouest 200 Ouest, Boul Dorchester Montréal, Quebec H2Z 1X4	Final
	14 Code de l'organisme parrain

15. Remarques additionnelles	16 Agent de projet — CDT
Parrainage conjoint avec le Groupe Interministériel pour la Recherche et le Developpement Enérgetique	J.H. Morgan

17. Résumé

Un moteur diesel monocylindre à préchambre a été utilisé pour la recherche sur le remplacement du combustible diesel par du propane. Le propane a été injecté en phase liquide sous forte pression directement dans la préchambre, à peu près au même moment que le combustible diesel dans un moteur diesel pur, l'allumage étant assuré par une bougie tenue en état d'incadescence, vu le très faible indice de cétane du propane. Les essais ont eu lieu à des taux de compression de 19:1, 22:1 et 25:1, et à des régimes de 1,200, 1,500 et 1,800 tr/min.

Au niveau égal d'opacité des fumées, la puissance du moteur à propane liquide a été de 15% supérieure au moteur diesel pur, les essais ayant montré l'importance de bien situer la bougie par rapport à l'injecteur pour obtenir un bon allumage et des délais d'inflammation courts. La disposition ultimement adoptée a donné de très bons résultats sur toute la plage d'utilisation du moteur au taux de compression de 25:1. En revanche, au deux autres taux, l'allumage n'a commencé à être fiable qu'à partir de quelque deux-tiers de la pleine charge. Au taux de compression de 25:1, et à pleine charge, la consommation spécifique observée a été de 11 MJ/kW×h dans le cas du moteur alimenté en diesel, de 9 MJ/kW×h pour le propane liquide et de 8 MJ/kW×h pour le propane en phase gaseuse (propane mélangé à l'air comburant et allumage par injection pilote).

Par des perfectionnements ultérieurs, le moteur au propane liquide à allumage par bougie "luisante" promet d'être non seulement faisable, mais aussi très utile.

18. Mots-clés	19. Diffusion
Propane Moteurs Diesel Injection	Le CDT dispose d'un nombre limité d'exemplaires.

20. Classification de sécurité (de ce rapport)	21. Classification de sécurité (de cette page)	22. Nombre de pages	23. Prix
Nonclassifié	Nonclassifié		

CDT/TDC 79-005
Rev 86

ACKNOWLEDGEMENTS

Financial support for this contract was made available through the Energy R & D Panel.

This support and the continuous guidance received through John. H. Morgan, Senior Development Officer responsible for the project, are gratefully acknowledged.

SUMMARY

This project used a single-cylinder prechamber diesel engine to investigate the use of propane as a diesel engine fuel. Two methods of delivering propane to the engine were evaluated. In the first method, fumigation, gaseous propane at ambient pressure was supplied to the engine's inlet manifold, where it was mixed with air entering the cylinder. Pilot injection of a small quantity of diesel fuel was used to provide ignition, since propane has a very low cetane number.

In the second propane fueling method, liquid propane was pumped to a high pressure and injected directly into the engine's prechamber at approximately the same time as diesel fuel would be injected in the original diesel engine. A continuously operating glow plug was used to provide ignition. The original equipment "Bosch" fuel injector was used to deliver propane, although the fuel injection pump and the fuel supply system were extensively modified to handle propane.

Tests were carried out at three compression ratios, 19:1, 22:1 and 25:1, and at three engine speeds, 1200, 1500 and 1800 rpm, first with the engine in its original diesel form and subsequently with the two propane fueling methods (fumigation and liquid propane injection).

The results showed that both propane fueling methods resulted in significantly less exhaust smoke than the original diesel at high loads. For the same level of smoke opacity, the liquid propane injection engine produced about 15% more power. The power output of the fumigated engine was knock limited to about the same level as that of the original diesel.

The tests with liquid propane injection revealed that the relative position of the glow plug and the fuel spray was critical to obtaining good ignition and short ignition delays. The final arrangement used provided excellent results throughout the engine's operating range at 25:1 compression ratio. However, at the two lower compression ratios, it was not possible to obtain reliable ignition at loads below about

two-thirds of full load. Another arrangement of glow plug and injector provided ignition at light loads with the 19:1 compression ratio, but had excessive rates of pressure rise at loads exceeding approximately 50% of fullload. It is believed that reliable ignition can be attained under all engine operating conditions with some additional development.

The liquid propane injection engine allowed complete substitution of propane for diesel fuel, while the propane fumigated engine required that as much as 25% of the total fuel energy be provided by diesel fuel to limit the maximum rate of pressure rise to less than 690 kPa/degree of crankshaft rotation, (100 psi/degree). Rates of pressure rise greater than this will result in excessive engine vibration and noise. It should be noted that even larger amounts of diesel fuel would be required in a fumigated direct injection engine. The high rate of mixing and subsequent rapid burning in the prechamber of this test engine reduce the susceptability of this particular engine to knock.

At 25:1 compression ratio, the energy consumption of the liquid propane injection engine was equal to or slightly lower than that of the original diesel. At lower compression ratios, however, the energy consumption of the liquid propane injection engine is higher than that of the original diesel. The same trend is maintained for fumigation. At lower compression ratios, the energy consumption falls with load until the fuel consumption of the propane fumigated engine is equivalent to that of the original diesel at full load. At higher compression ratios, the energy consumption is not very sensitive to load and is significantly better than that of the original diesel. At 25:1 compression ratio, the full load indicated specific fuel consumptions were 11 MJ/kW×h, 9 MJ/kW×h and 8 MJ/kW×h for diesel, liquid propane injection and propane fumigation respectively. Thus, the fuel consumption is improved 18% with liquid propane injection and 27% with propane fumigation.

In general, NO_x emissions of the liquid propane injection engine are lower than those of the original diesel at light loads and decrease further as load is increased. With fumigation, the NO_x emissions are very low at light loads but increase sharply as full load is approached, especially at low compression ratios. For both propane fueling systems, the influence of load was reduced by increased compression ratio. The tradeoff for the advantages of liquid propane injection is the increased mechanical complexity of the system. Additional development is needed to provide a simpler and more reliable engine fueling system. A number of problem areas have been identified in this initial project which should be addressed in subsequent work. This project has, however, demonstrated the feasibility and promise of the liquid propane injection engine with glow plug ignition.

SOMMAIRE

A fin de mener la recherche sur le remplacement du combustible diesel par du propane, un moteur diesel monocylindre à préchambre a été utilisé. Deux méthodes d'alimentation distinctes ont été évaluées. Avec la première méthode, le propane en phase gazeuse est délivré à la pression atmosphérique dans la tubulure d'admission où il est mélangé à l'air comburant allant vers le cylindre. L'injection pilote d'une petite quantité de combustible diesel précédait l'injection principale dans le but d'amorcer l'allumage, étant donné le très faible indice de cétane du propane.

Avec la deuxième méthode d'alimentation, le propane est injecté en phase liquide, et sous forte pression, directement dans la préchambre, à peu près au même moment que le combustible diesel dans un moteur diesel pur, l'allumage étant assuré par un bougie tenue constamment en état d'incadescence. Le système d'injection d'origine, fabriqué par Bosch, a été conservé, mais la pompe et le circuit d'alimentation ont été profondément modifiés.

Les essais ont eu lieu à des taux de compression de 19:1, 22:1 et 25:1 et à des régimes de 1,200, 1,500 et 1,800 tr/min, d'abord avec le moteur diesel pur et ensuite avec le même moteur mais alimenté selon l'une et l'autre des deux méthodes décrites plus haut.

L'émission de fumées a été considérablement moins forte avec le propane sous ses deux formes que celle du moteur diesel aux charges élevées. A niveau égal d'opacité des fumées, la puissance du moteur au propane liquide a été d'environ 15 p. 100 supérieure, tandis que pour le moteur alimenté en propane gazeux, elle est restée sensiblement égale à celle du moteur diesel pur à cause des anomalies de combustion (cognement).

Les essais sur le moteur au propane liquide ont montré l'importance de bien situer la bougie par rapport à l'injecteur pour obtenir un bon allumage et des délais d'inflammation

courts. La disposition qui a été ultimement adoptée a donné de très bons résultats sur toute la plage d'utilisation du moteur au taux de compression de 25:1. En revanche, aux deux autres taux, l'allumage n'a commencé à être fiable qu'à partir des quelque deux-tiers de la pleine charge. A une certaine disposition relative de la bougie et de l'injecteur, l'allumage a pu être amorcé à faible charge au taux de 19:1, mais au prix d'une montée en pression de plus en plus forte, devenant excessive lorsque la charge dépassait 50 p. 100 environ. On croit cependant pouvoir obtenir un allumage fiable sur toute la plage d'utilisation du moteur grâce à des perfectionnements techniques.

Le propane liquide permet une substitution totale du combustible diesel alors qu'avec le propane en phase gazeuse, if faut compter sur un apport diesel pouvant atteindre 25 p. 100 de la quantité totale de combustible, dans le but de garder le taux de montée en pression maximal dans les limites de 690 kPa (100 lb/po^2/degré) par degré de rotation du vilebrequin, une montée en pression plus rapide risquant d'entraîner des niveaux de vibration et de bruit excessifs. Il est intéressant d'observer que, dans le cas du propane en phase gazeuse à injection directe, la proportion de combustible diesel d'apport est encore plus grande. Par contre, le moteur d'essai a montré une moins grande susceptibilité au cognement grâce à l'obtention d'un mélange très homogène dans la préchambre, ce qui favorise une combustion plus rapide.

Au taux de 25:1, la consommation du moteur au propane liquide est égale ou légèrement inférieure à celle du moteur diesel pur, mais elle augmente par rapport à ce dernier aux taux de compression plus faibles. Dans le cas du moteur au propane en phase gazeuse, les mêmes tendances ont été constatées. Aux taux de compression faibles, la consommation diminue proportionnellement à la charge pour atteindre celle du moteur diesel pur à pleine charge, dans le cas de l'alimentation en phase gazeuse. A mesure que le taux de compression augmente, la consommation cesse graduellement de

dépendre de la charge et devient nettement meilleure que dans le cas du diesel pur. Ainsi, au taux de 25:1 et à pleine charge, la consommation spécifique observée a été de 11, 9 et 8 MJ/kW×h dans le cas du moteur alimenté en diesel, propane liquide et propane en phase gazeuse, respectivement. On dégage de la sorte un gain de 18 p. 100 pour l'alimentation en phase liquide et de 27 p. 100 pour l'alimentation en phase gazeuse.

En ce qui concerne les émissions de NO_x , dans le cas du moteur au propane liquide, elles sont plus faibles que celles du moteur diesel pur aux faibles charges et diminuent à mesure que la charge augmente. Dans le cas du moteur au propane en phase gazeuse, elles sont très faibles aux faibles charges, mais augmentent radicalement à l'approche de la pleine charge, surtout lorsque le taux de compression est faible. Dans les deux cas, cependant, on corrige l'influence de la charge en relevant les taux de compression.

En contrepartie des avantages offerts par le propane liquide, on doit accepter une complexité mécanique plus grande; d'autre part, de plus amples recherches seront nécessaires dans le but de redessiner le circuit d'alimentation pour le rendre plus fiable. La recherche a mis en évidence certains points faibles qui pourront être résolus par des travaux ultérieurs. Elle a néanmoins montré que le moteur au propane liquide à allumage par bougie "luisante" est non seulement faisable, mais aussi très prometteur.

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1. INTRODUCTION

Propane has been the most successful of potential alternative fuels for transportation in Canada. Its primary attraction is reduced cost (due to the absence of motor fuel taxes), but unlike most other alternative fuels, vehicle range and drivability with propane are nearly equivalent to that with gasoline.

With increasing production of natural gas in Canada, the supply of propane should increase modestly as well. This increase will be partially offset by the loss of propane output from inactivated oil refineries. While the amount of propane produced will never be able to meet all of Canada's transportation fuel needs, a steadily growing supply will enable propane to be substituted for gasoline in a greater number of applications.

Propane is widely distributed geographically, both in motor vehicle quality and in commercial quality for non-engine applications. The number of vehicle refueling stations in a given area is still small, however. Thus, propane is best suited for use in fleet vehicles which typically are refueled at a central facility. Equipment to convert the spark-ignition gasoline-fueled engines in light and medium duty vehicles to use propane is widely available, although investigations carried out for the Ontario Ministry of Transportation and Communications (MTC) by the Engine Research and Development Laboratory (ERDL) of the University of Toronto in the Department of Mechanical Engineering (UTME) have shown that currently available equipment does not provide optimum performance and economy (1). Despite the availability of this equipment, there is still a sizeable segment of the total fleet market which cannot presently be converted to propane. Virtually all heavy-duty vehicles and an increasing number of medium and light duty vehicles are powered by diesel engines, for which no propane conversion equipment is available. The

research work performed in this project has demonstrated the feasability of successfully operating diesel cycle engines on propane.

There are three potential methods of using propane in diesel engines. In the first method, propane is mixed with air and the resulting mixture introduced into the engine cylinders, a technique known as fumigation. Ignition is provided by the injection of a small quantity of diesel fuel in the normal manner (pilot injection). The engine is therefore a dual-fuel engine. The proportions of propane and diesel fuel must be carefully controlled if excessive rates of pressure rise during the combustion process are to be avoided (2,3).

Fumigation is effective in constant speed engine applications such as generating sets, because the engine's governor decreases the amount of diesel fuel injected to compensate for the propane inducted with the air. Fumigated vehicle engines have a mechanical linkage to control the ratio between propane and diesel fuel. This arrangement is a compromise, however, as it provides a constant percentage of propane while the amount of propane that can be tolerated by the engine varies with operating conditions. In addition, the linkage may be tampered with to provide more fuel (and therefore more power), to the detriment of engine durability and pollutant emission levels. Finally, the amount of propane that can be substituted for diesel fuel is limited to about 30-40% in most fumigated engines. Thus, despite its simplicity, propane fumigation is not a particularly effective fueling method for transportation applications.

Another dual-fuel possibility is to inject liquid propane into the engine, along with a small pilot injection of diesel fuel to provide ignition. The ignition source is necessary because propane has a very low cetane number. This concept has been explored in a parallel investigation in cooperation with SNC, Inc. of Montreal. The final report for the SNC project should be consulted for further details (4).

Propane may also be used as the sole engine fuel. Liquid propane, rather than diesel fuel, would be injected into the engine. Again, some means of ignition, in this case either a continuously operating glow plug or a timed spark, must be provided. Thus, the propane-fueled diesel would effectively be a stratified charge engine. A continuously operating glow plug was chosen to be the ignition source in this research program due to its simplicity.

Because propane is stored as a compressed liquid, significant modifications to the fuel supply system and the fuel injection pump and injector were necessary to keep the propane liquified and to accommodate propane's low lubricity. The following sections of this report detail the modifications made and present the results of an extensive test program to evaluate the feasability of operating diesel engines on injected propane with glow plug ignition assist.

2. EXPERIMENTAL APPARATUS

2.1 TEST ENGINE

"WAUKESHA" CFR (Cooperative Fuel Research) one-cylinder variable compression indirect injection (IDI) diesel S/N 791712.

Bore:	3.25 in (82.55 mm.)
Stroke:	4.50 in (114.30 mm.)
Displacement:	37.33 in^3 (611.73 cm^3)
Antechamber:	swirl combustion type
bore:	1.625 in (41.275 mm.)
length:	adjustable

Figure 2-1 shows the variable compression head and the cylinder of the CFR engine. Compression ratio is varied by screwing an expansible plug in or out of the cylindrical combustion chamber. For this project, the variable compression ratio plug was replaced with a plug of fixed dimensions incorporating a mounting hole for a pressure transducer (Figure 2-2). Compression ratios of 19:1, 22:1 and 25:1 could be selected by fitting shims between the plug and the mounting face ofthe cylinder head. This freed the threaded access hole at the top of the combustion chamber, normally used to mount a knock sensor, for installation of a glow plug and/or injector.

Three different fuel delivery arrangements were used. For baseline tests with diesel fuel, the "American Bosch" injection pump and injector standard on the CFR cetane engine were used. This system also served to deliver the pilot injection of diesel fuel during the fumigation tests. Gaseous propane was supplied to the engine during the fumigation tests through a 1/4 inch diameter tube drilled with opposed holes and placed across the air inlet pipe. A two stage regulator-vaporizer and a single stage final regulator reduced the propane pressure to approximately 2 kPa (0.3 psi).

The fuel system was changed completely for the tests with liquid propane injection. Due to the fact that propane is a gas at atmospheric temperature and pressure, fuel system components

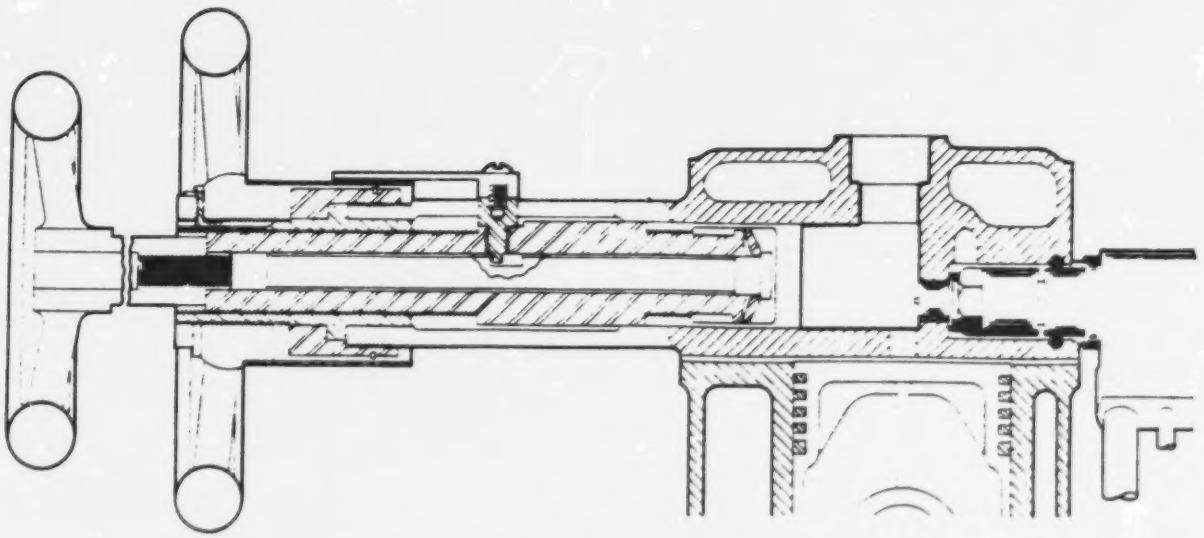


Figure 2.1 Variable compression head and cylinder of CFR cetane engine

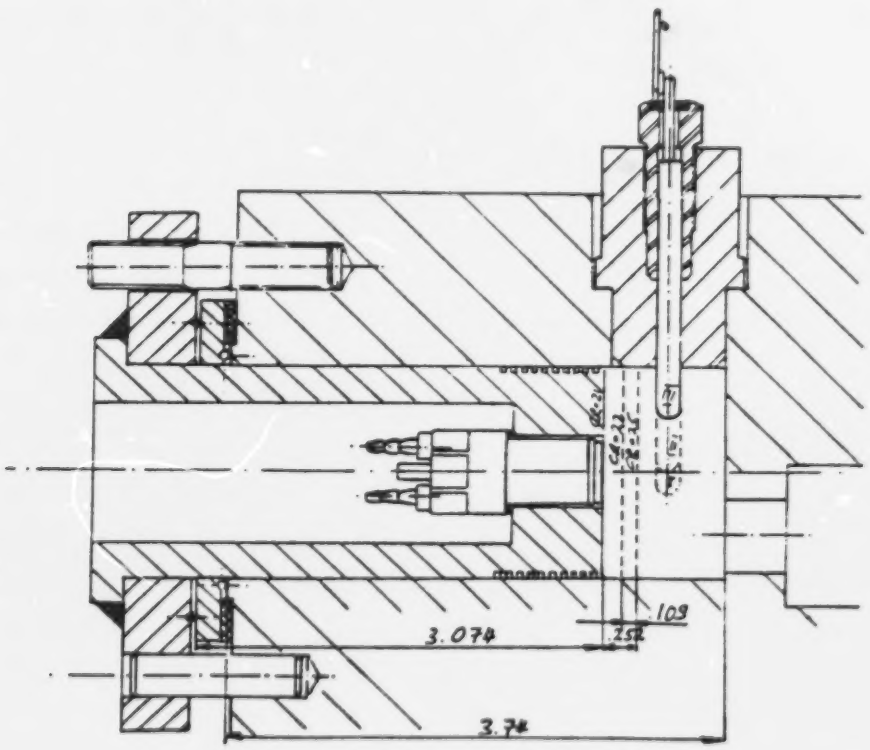
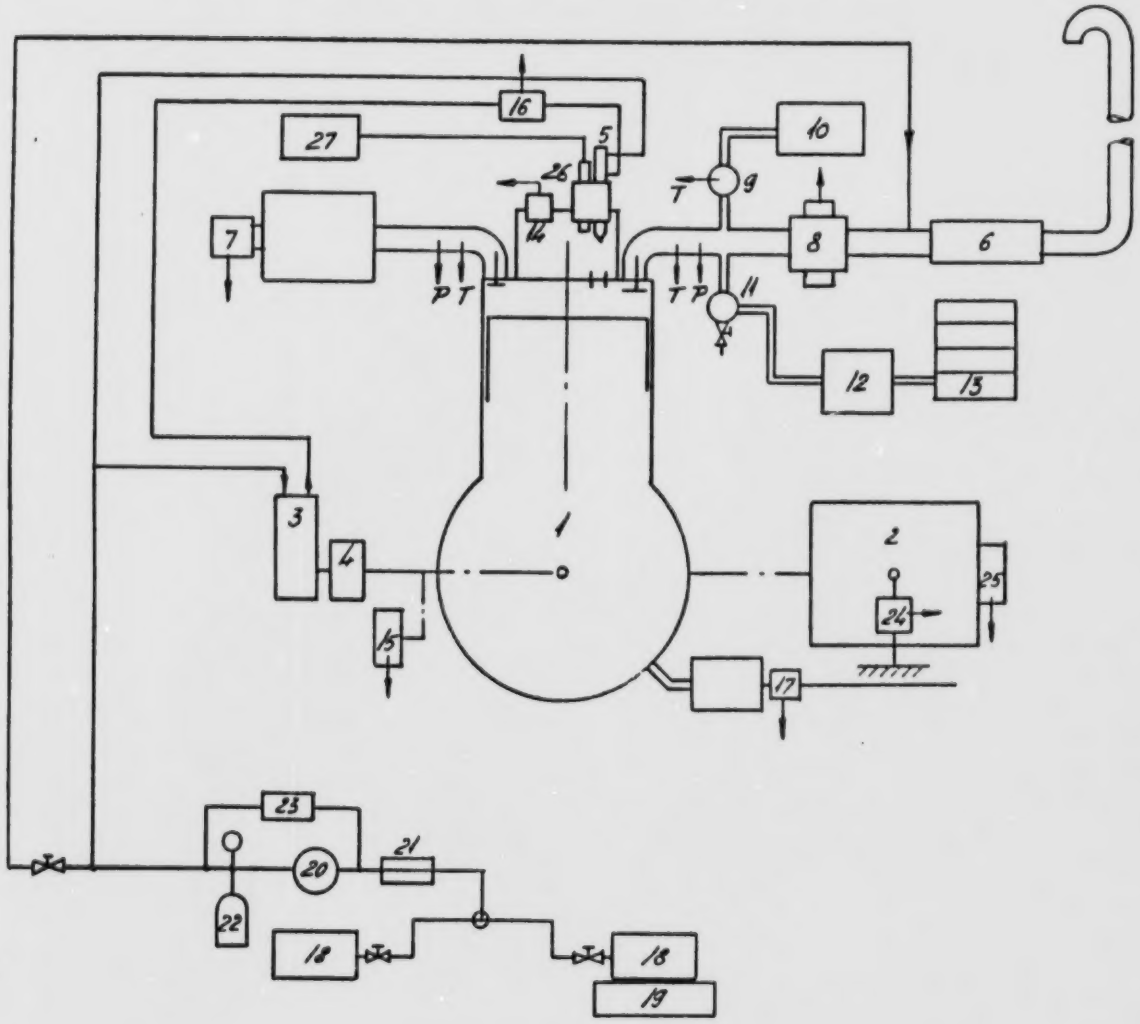


Figure 2.2 Fixed compression ratio swirl chamber plug with pressure transducer

in addition to those normally present on a diesel engine were necessary to keep the propane liquified. A block diagram of these components is included in Figure 2.3. Propane from the liquid takeoff of the tank was first filtered using a Matheson 6184 replaceable element high pressure filter (100% efficient to 0.2 microns) to remove dirt and debris (Extensive laboratory test experience with propane has shown that even HD-5 propane is very dirty - excellent filters are a must). A diaphragm pump ("American Lewa" model EKM-1 with 16 mm size M510S head), driven by an explosion proof 110 VAC electric motor was used to pressurize the liquid propane drawn from the tank to 4.2 to 5.2 bars (600-750 psi). This pressure is above the saturation pressure for any temperature likely encountered in the fuel supply system and should therefore ensure that the propane stays liquified. The pressure is maintained by a relief valve ("Nupro" 6S 4R3A) which recirculates propane as necessary to the pump inlet. An accumulator ("Greerolater" 800 201) was used to dampen pulses from the supply pump. Figure 2.4 shows the primary pressure supply system.

The standard fuel injection pump was replaced by two "Robert Bosch" fuel injection pumps (PFR 1K80A 427/11), one for the pilot injection of diesel fuel and one for liquid propane injection, although pilot injection was not used in this project. The camshafts which actuated these pumps were connected in series and driven from the front end of the engine camshaft as shown in Figure 2.5. An "American Bosch" Model TMB 12A2 variable timing device was incorporated between the drive and the first pump (diesel fuel) to allow setting the diesel pilot injection timing. A "Candy Manufacturing Company" Model 1A Dynamic Differential was used to drive the second pump (liquid propane) from the first so that the phase angle between the two pumps could be varied. This allowed setting the liquid propane injection timing to a different value than the diesel pilot injection. Pressurized propane from the primary supply system was fed to the second pump. Extensive modifications to the barrel and plunger (6 mm diameter #1418405002-800) were necessary to ensure the delivery of propane with minimum leakage and minimum wear despite the low



1 - engine
2 - dynamometer
3 - liquid propane injection pump
4 - angular phase shifting mechanism
5 - injector
6 - muffler
7 - inlet air flowmeter and pulsati damper
8 - smokemeter
9 - heated filter and lines
10 - heated HC analyzer
11 - filter and drain
12 - exhaust sample conditioner
13 - NO, O_2, CO and CO_2 analzyers
14 - cylinder pressure transducer
15 - crank angle encoder
16 - propane line pressure transducer
17 - blow-by flowmeter and pulsati damper
18 - propane cylinders
19 - balance
20 - pump
21 - filter
22 - accumulator
23 - pressure check valve
24 - load cell
25 - tachometer
26 - glow plug
27 - electric power supply
T - temperature measuring point
P - pressure measuring point

Figure 2.3 Overall layout of test equipment and instrumentation

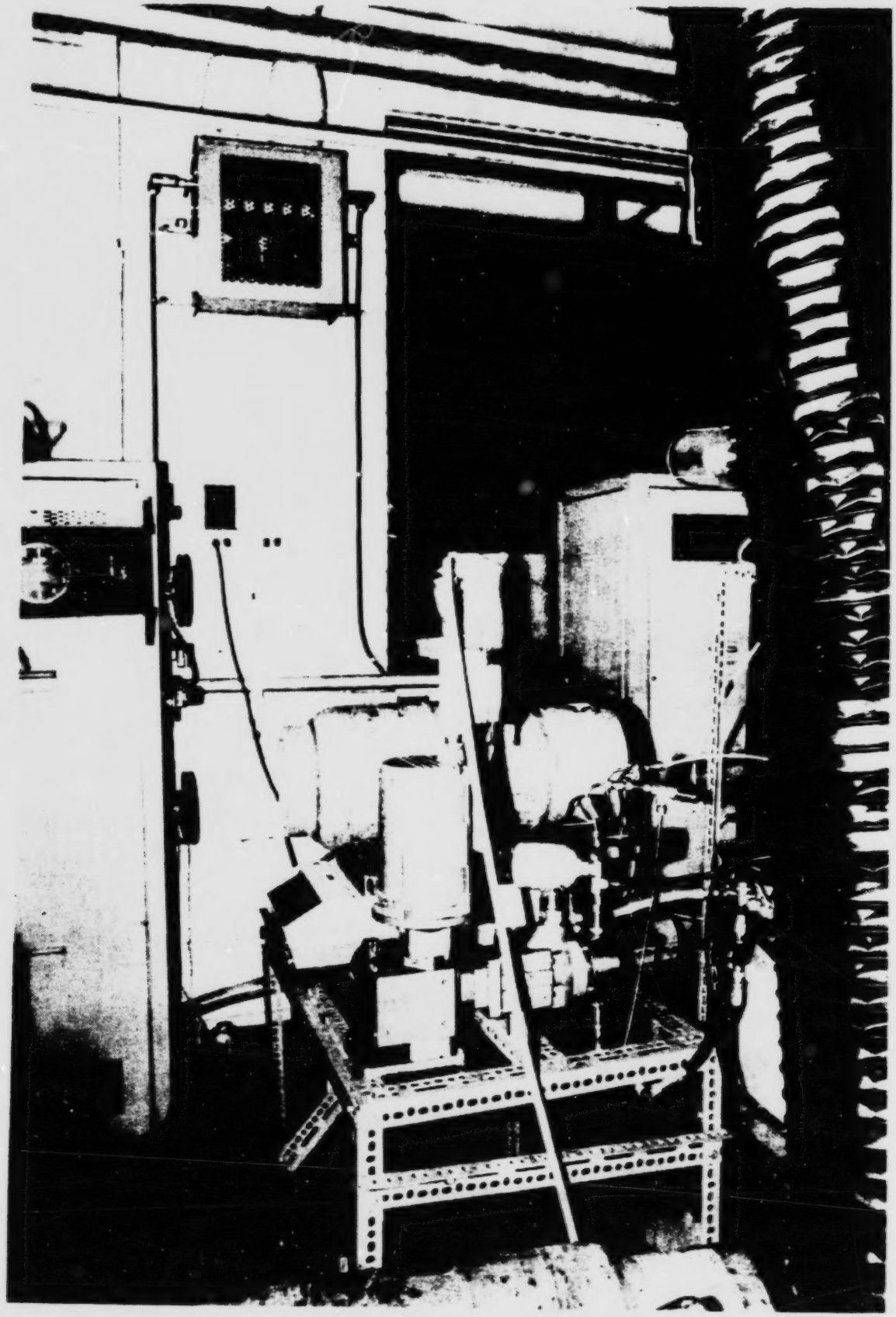


Figure 2.4 Liquid propane consumption measurement and primary supply system

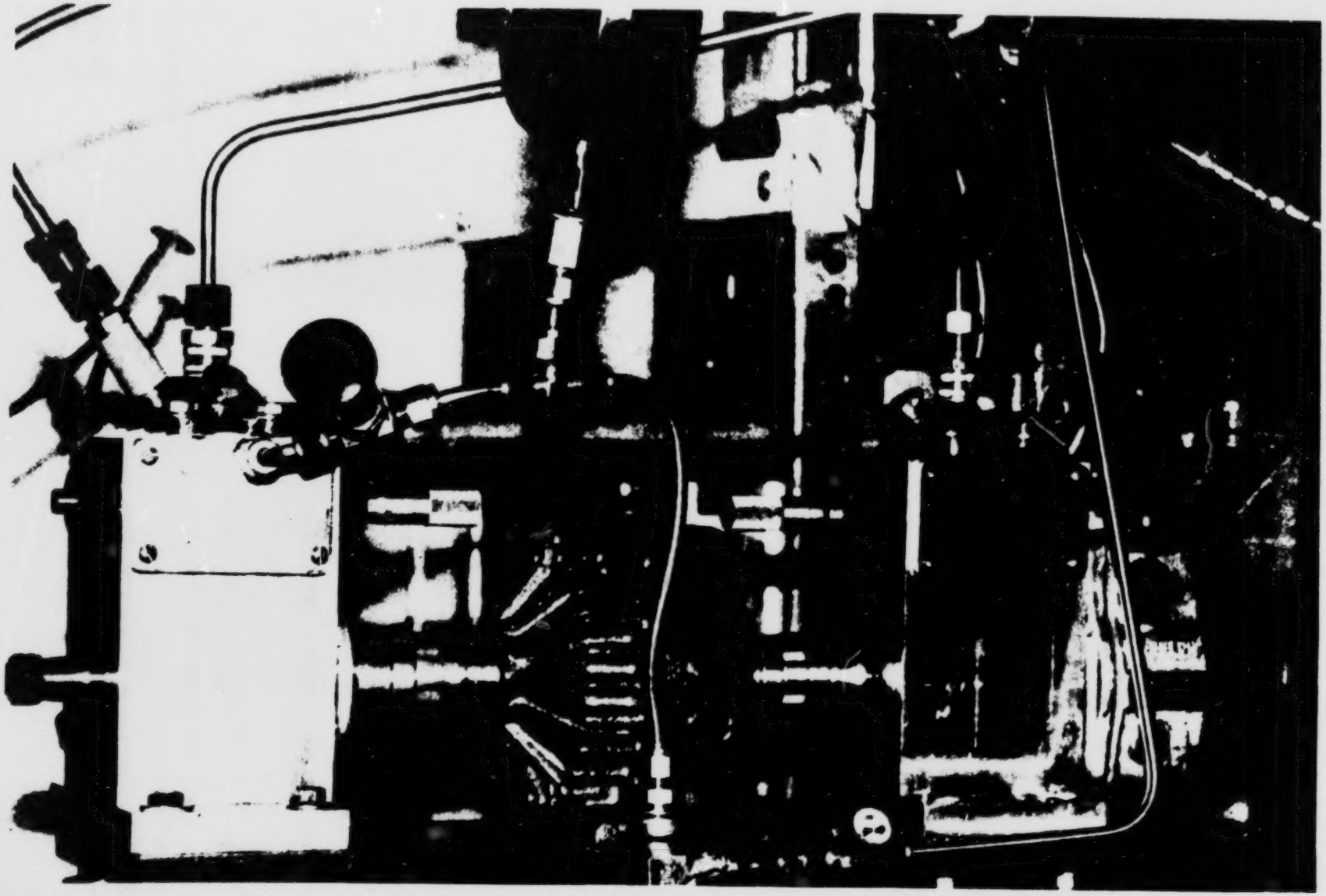


Figure 2.5 Detail of liquid propane injection pump (left side-the diesel fuel pump on the right is not used in this project). Note phase control, micrometer for rack control and supplementary lubrication of pump

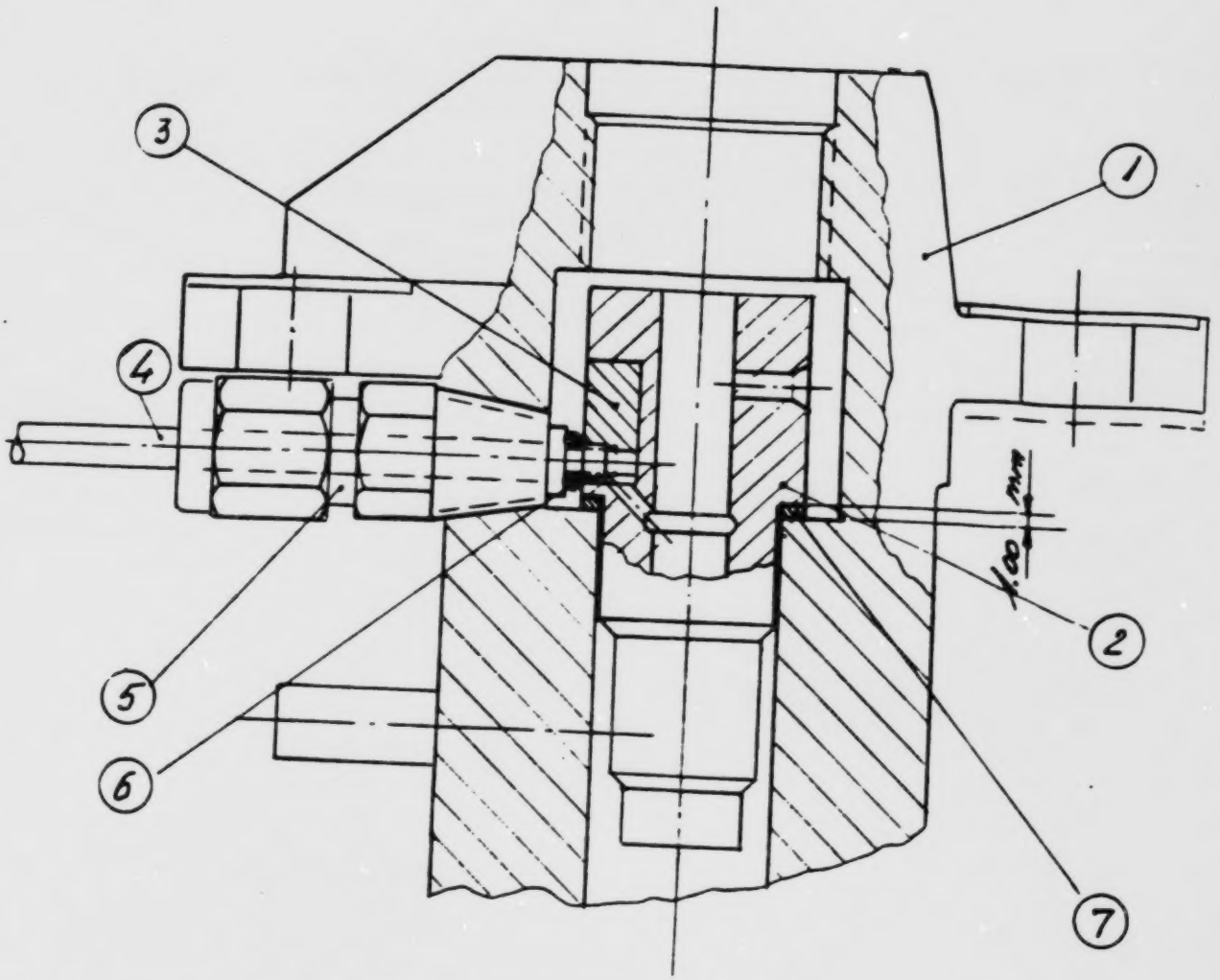
lubricity of propane compared to diesel fuel. Referring to Figure 2.6, a copper gasket was added under the barrel shoulder to avoid leaks from the fuel supply chamber to the lower housing containing the camshaft and roller tappet. This was necessary because the fuel inlet pressure is nearly 100 times that used with diesel fuel. The groove and inclined channel built into the barrel to return the diesel fuel high-pressure leak-off to the supply side was modified to supply lubrication oil to the barrel and plunger assembly. The barrel was machined to accept a mild steel insert (#3 in Figure 2-6) containing a 1/8 inch diameter oil supply tube (#4). The insert was glued to the barrel with epoxy to avoid welding or soldering. Heat from either of the latter would have destroyed the precision fit of the plunger and barrel assembly. The oil supply tube was held in place by a Swagelok male connector (#5). The lower end of the connector compressed an O-ring (#6) which sealed the oil supply tube to the barrel.

Oil for sealing and lubrication was provided by a cylinder pressurized by propane at 4.1-5.6 bars (600-750 psi) from the supply pump (this system is shown in Figure 2.5 but not 2.3). A check valve was necessary to prevent reverse flow when the injection pump was delivering fuel to the injector. A manual shutoff valve was also placed in the line. Oil flow was shut off during tests as it was discovered that enough lube oil was getting into the propane to cause ignition without the glow plug (albeit with considerable smoke). The oil flow was turned on during warm-up to minimize plunger and barrel wear.

Figures 2.7 to 2.9 present details of the glow plugs used in different stages of the project.

2.2 TEST EQUIPMENT

The overall layout of test equipment and instrumentation is shown schematically in Figure 2-3 and pictorially in Figure 2.10. The following sections present the specifications for the test equipment and instrumentation used.



1 - Injection pump "Bosch" model PFR 1K80A 427/11
2 - 6 mm diameter barrel, "Bosch" part number 1418405002-800
3 - Mild steel insert
4 - Oil supply tube
5 - Fitting "Swagelock" male connector SS-100-2-2
6 - O-ring
7 - Copper gasket

Figure 2.6 Modified fuel injection pump for liquid propane delivery

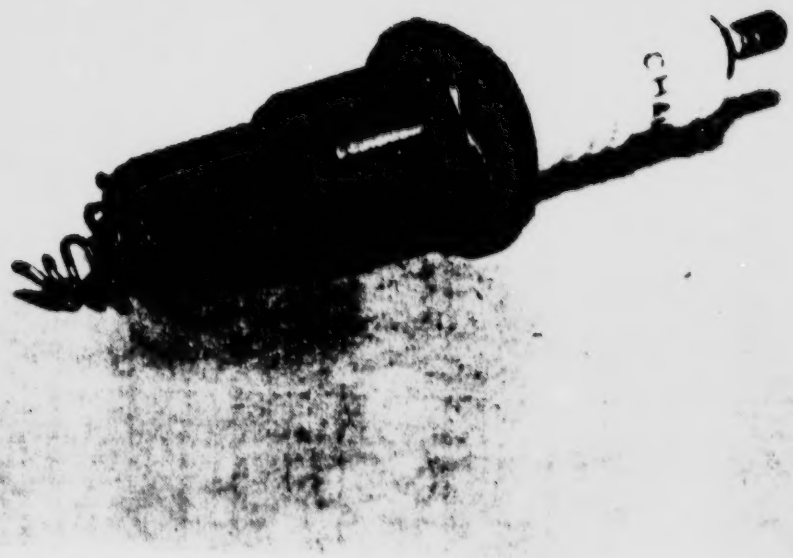


Figure 2.7 Platinum glow plug

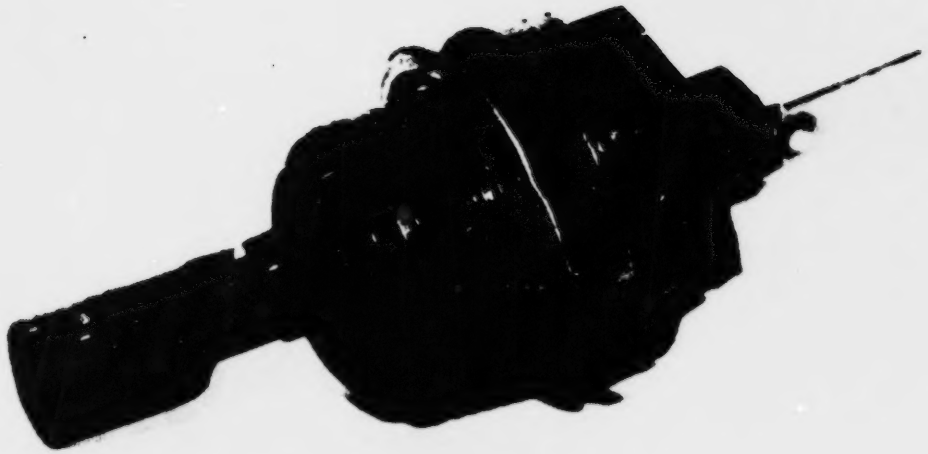


Figure 2.8 Shrouded glow plug

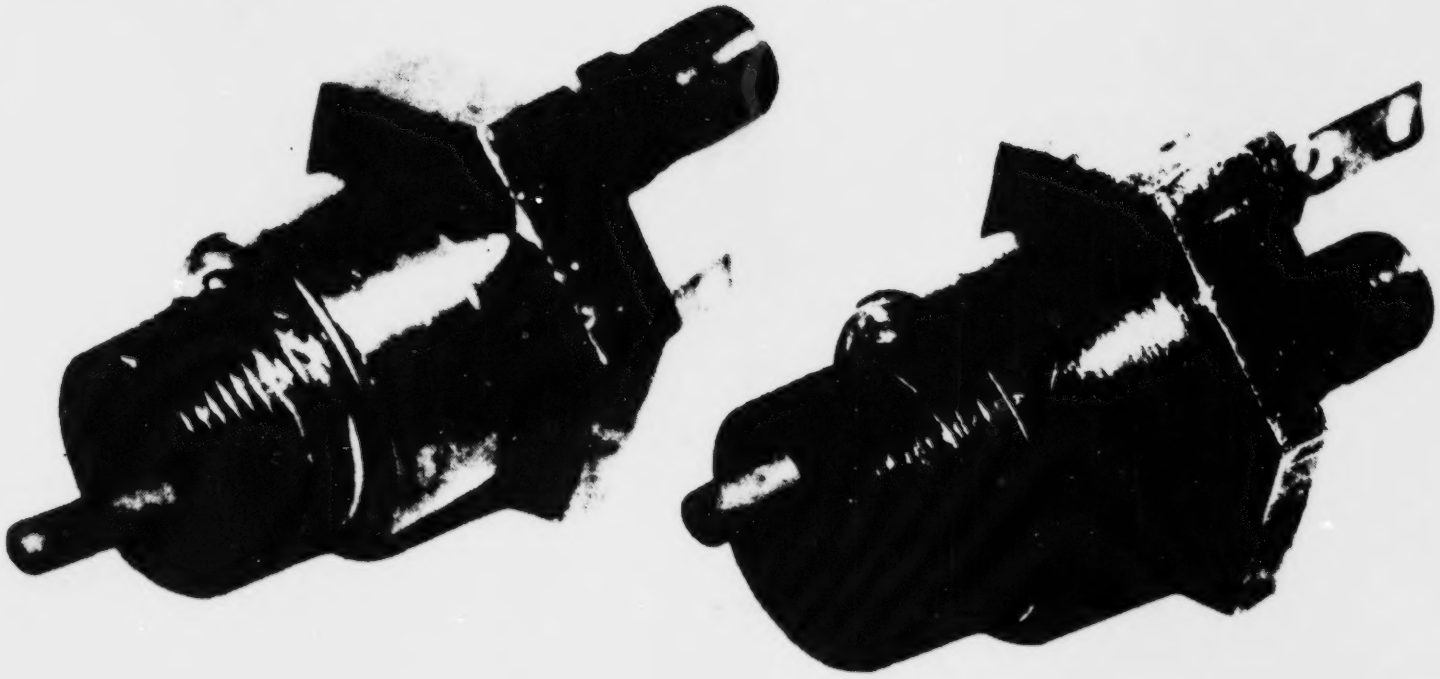


Figure 2.9 Glow plug and poppet injector in one body (shrouded and exposed)

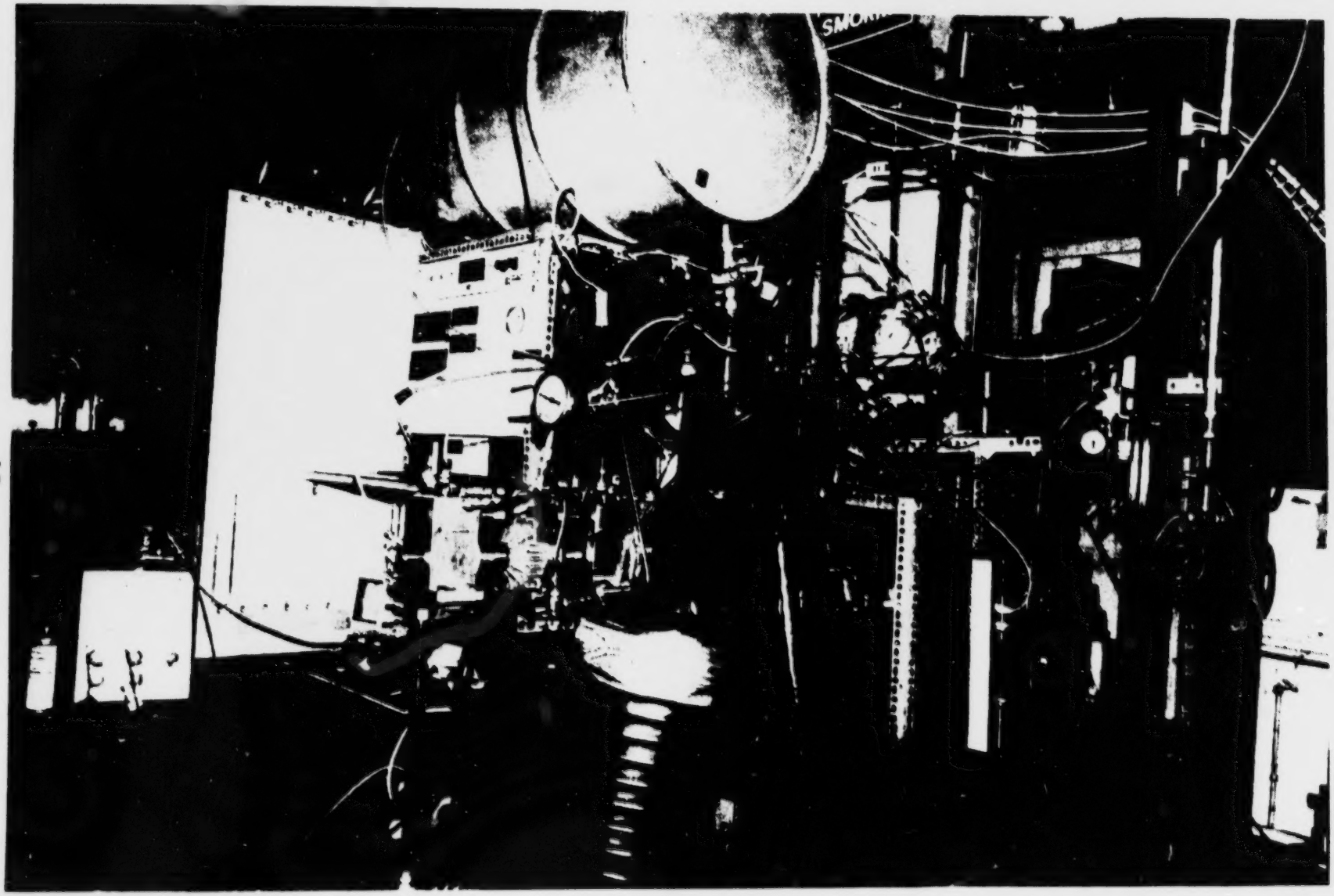


Figure 2.10 Overall view of the experimental equipment and instrumentation

Fuel consumption measurements were made on a mass basis. Two fuel tanks were used for each fuel, a service tank and a weigh tank, the latter supported by an electronic balance. After recording the mass of the weigh tank, activation of an electronic timer switched fuel delivery from the service tank to the weigh tank for a preset interval. The final weight of the weigh tank could then be determined when fuel delivery was switched back to the service tank.

2.2.1 Smoke Measurement

"CELESCO" model 107 in-line exhaust smokemeter

Measuring method:	attenuation of the intensity of a collimated light beam by smoke aerosol absorption and scattering	
Scales	opacity or smoke density (K)	
Opacity range:	0 to 100%	
	Resolution:	0.1%
	Accuracy:	±0.5%
Smoke density range:	0 to 20 m^{-1}	
	Resolution:	0.01 m^{-1}
	Accuracy:	0.1 m^{-1} for $\leq$ 2 m^{-1}
		±5% of reading > 2 m^{-1}

2.2.2 Temperature Measurement

"DORIC" TRENDICATOR series 400A with 10 channels switching box and "OMEGA" thermocouples type K (Chromel-Alumel)

Range:	-200 to 1,250°C
Accuracy of thermocouple:	±2.2°C or 0.75%, whichever is greater, for $\geq$ 0°C
Accuracy of indicator:	±0.5°C
Junction:	grounded

2.2.3 Pressure Measurement

2.2.3.1 Cylinder pressure

- "AVL" type 12QP505 c1 radiation shielded, water cooled quartz transducer

  Measuring range: 0-150 bar dynamic load
  Temperature range: $\leq$ 240°C
  Sensitivity: in 34 pC/bar range, individually calibrated
  Linearity: < ±1%
  Temperature effect: 0.01% FS/°C
  Natural Frequency: 65 kHz

- "AVL" model 3059B-A02.1 charge amplifier

  Sensitivity ranges: 10^{-5} to 1 V/pC
  Frequency response: near DC to 100 kHz (-3 dB)
  Linearity error: <0.05%
  Temperature drift: 0.2 to 2 mV/°C
  Operating temperature range: 0 to 60°C

2.2.3.2 Fuel line pressure

- "AVL" type 31 DP strain-gauge transducer

  Construction: full bridge, 120 ohms
  Measuring range: 500 kg/cm^2
  Non-linearity: <0.5%
  Operating temperature range: -10 to 90°C
  Temperature influence on zero or sensitivity: <0.04%/°C
  Natural frequency: approximately 50 kHz

- "AVL" type 3009-A01 strain-gauge amplifier

  Bridge supply voltage drift: <0.003%/°C
  Bridge voltage accuracy: ±0.05%
  Output frequency response: 0 to 70 kHz flat, unfiltered
  Linearity: 0.03% FS
  Zero drift: <1 µV/°C
  Operating temperature: 0 to 50°C

2.2.3.3 Inlet air, exhaust, blowby pressures: U-tube mercury and water manometers

2.2.7 Injector Needle Lift Measurement

- "WOLFF"-ANALOG SENSOR integrated Hall effect transducer

Supply voltage:	maximum 16 V DC
Magnetic flux density:	no limit
Sensitivity:	1 V/mm
Frequency response:	25 kHz (-3 dB)
Thermal drift:	-3 mV/kGauss-^{o}C
Operating temperature:	-40 to 150^{o}C

- UTME (University of Toronto, Mechanical Engineering) instrumentation amplifier

For details of installation of the transducer on the "STANADYNE" injector, see Figure 2.11.

2.2.8 Air Rate of Flow

"TSI" model 2013 with air linearizer card and indicator model 67

Accuracy:	±2% reading ±0.2% FS
Repeatability:	±0.1% reading ±0.05% FS
Response time:	0.1 s
Temperature compensation:	0 to 40^{o}C
Operating pressure:	100 ± 20 kPa

2.2.9 Emissions

2.2.9.1 Hydrocarbons analyzer

- "BECKMAN" model 404

Measuring method:	flame ionization method
Sample treatment:	heated filter, supply line and analyzer, adjustable from 30 to 205^{o}C (set at 191^{o}C)
Ranges:	10, 25, 100, 250 ppm, 0.1, 0.25, 1.0% hydrocarbons as methane

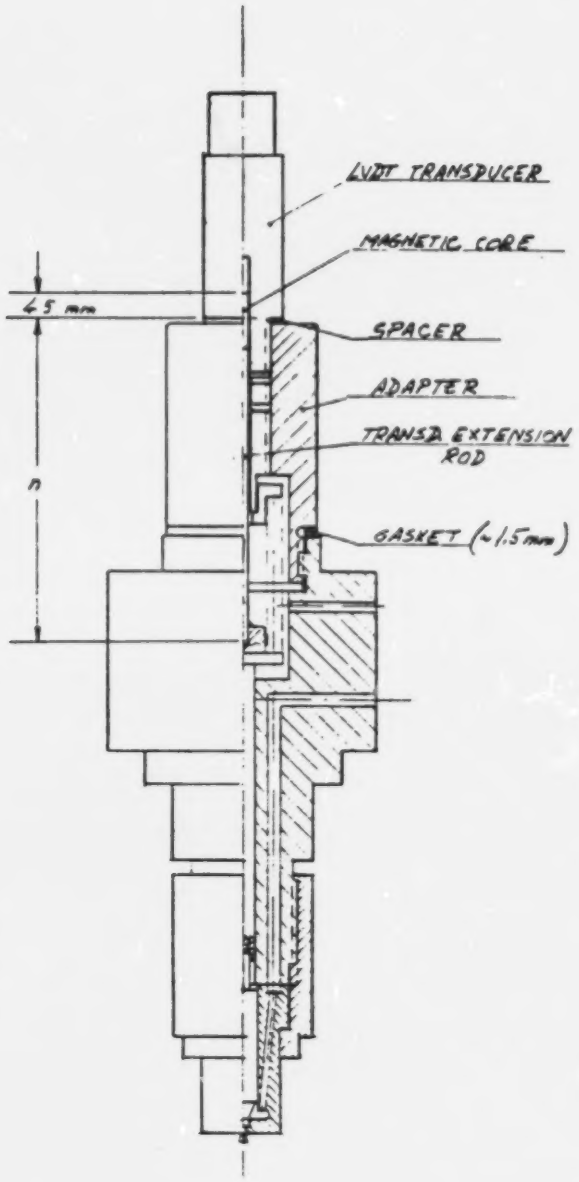


Figure 2.11 Needle lift transducer installed on BOSCH injector

Response time:	<1.5 s detection module 5.5 s complete system (48" sample line) for 95% of final reading

2.2.9.2 NO analyzer

- "BECKMAN" model 951

Measuring method:	chemiluminiscence
Ranges:	10, 25, 100, 250, 1000, 2500, 10,000 ppm
Sensitivity:	0.1 ppm on 10 ppm range
Linearity:	±1% FS
Precision:	0.5% FS
Dynamic response:	≤1 s

2.2.9.3 O_2 analyzer

- "BECKMAN" model OM-11

Measuring method:	polarographic
Range:	25%
Accuracy:	±1% FS
Linearity:	±1.2% O_2
Response speed:	90% within 300 to 600 ms

2.2.9.4 CO and CO_2 analyzers

- "BECKMAN" model 864

Measuring method:	non-dispersive infrared (NDIR)
Ranges:	2, 10% CO 20% CO_2
Precision:	1% FS
Noise:	1% FS
Zero or span drift:	±1% FS for 24 h
Response speed:	adjustable, 90% in 0.5 to 26 s

2.2.9.5 Exhaust sample conditioning system for NO, O_2, CO and CO_2 analyzers

- "BECKMAN", with diaphragm compressor and air dryer.

2.2.10 Crank Angle Encoder

"AVL" model 360 C/600/S optical infrared encoder

Trigger pulse:	1 low pulse/revolution, can be aligned with top dead centre or other required crank shaft event
Crank degree marks:	maximum 3,600 signal edges/revolution (0.1 crank angle resolution)
Speed range:	250 to 8,000 rpm
Temperature range:	0 to 60°C
Accuracy:	±0.05° crank angle

2.2.11 Dynamic Display and Measurement (for cylinder and fuel line pressures, injector needle lift and crank angle marks)

2.2.11.1 Four channel storage oscilloscope

- "TEKTRONIX" model 5103N system

Calibrated deflection:	1 mV to 5V/division
Bandwidth:	DC to 1 MHz (usable)
Accuracy:	within 2%
Time base:	1 µs to 5 s/division
Time base accuracy:	within 3 to 5%

2.2.11.2 Two channel oscilloscope

- "B+K PRECISION" model 1476

Calibrated deflection:	10 mV to 20 V/division
Bandwidth:	DC to 10 MHz (-3 dB)
Accuracy:	±5%
Time base:	1 µs to 0.5 s/division
Time base accuracy:	±5%

2.2.12 dP/dt Adapter

UTME construction. Supplied with the output of the cylinder pressure amplifier, the output is the differential of this signal.

3. EXPERIMENTAL PROCEDURE

During the test, the following parameters are recorded:

- engine speed - rpm
- engine torque - N.m
- engine friction torque - N.m (at stabilized working temperature, using the dynamometer motoring capability)
- mass fuel consumption - g
- test duration - s
- inlet air flow - standard L/min.
- inlet pressure drop - in.Hg
- exhaust back pressure - in.Hg
- blow-by differential pressure - in.Hg (to determine flow)
- inlet air temperature - deg.C
- exhaust temperature at engine - deg.C
- oil temperature - deg.C
- water temperature - deg.C
- smoke opacity - %
- HC emission as CH_4 - ppm
- NO/NO_x emission - ppm
- O_2 emission - %
- CO emission - %
- CO_2 emission - %
- cylinder pressure - bar (x 100 = kPa)
- dP/dt - bar/deg
- oil pressure - kPa
- fuel line pressure - bar
- beginning of injection - deg BTDC
- end of injection - deg ATDC
- maximum needle lift - mm
- combustion start - deg BTDC
- atmospheric pressure - kPa
- atmospheric vapour pressure - kPa

From those values, a number of derived parameters are determined through computation:

- ignition delay - deg
- standard atmospheric conditions correction factor
- fuel-air ratio
- corrected indicated power - kW
- corrected brake power - kW
- indicated specific fuel consumption - MJ/KW.h
- brake specific fuel consumption - MJ/kW.h
- indicated mean effective pressure - bar
- brake mean effective pressure - bar
- indicated efficiency
- brake efficiency

After setting of desired compression ratio and calibration of instrumentation, the engine-dynamometer system is maintained at the desired speed by control of dynamometer load and fuel rack position.

All data are recorded under stabilized thermal conditions.

4. RESULTS AND DISCUSSION

In this section test results for three methods of operating the engine will be compared. These are the original diesel fuel injection, propane fumigation with diesel pilot injection and operation on propane only using glow plug ignition. The glow plug is necessary because propane has a very low cetane number. The data presented in all cases was with the injection timing resulting in best power output, limited to a maximum rate of pressure rise of less than 690 kPa/degree of crankshaft rotation (100 psi/degree). Higher rates of pressure rise resulted in excessive engine vibration and noise.

The original diesel fuel injectior was used to deliver liquid propane to the engine although the fuel injection pump and fuel supply system were extensively modified as stated in Chapter 2 (Experimental Apparatus).

In the propane fumigation tests the minimum amount of diesel pilot injection necessary to ensure stable operation was used. The percentage of the total fuel energy supplied by the diesel fuel therefore varies from test to test. This data will be presented subsequently.

The power output of diesel engines is normally limited by the level of smoke emissions that can be tolerated. Figures 4.1 through A.9 (see Appendix) show the smoke emissions of the test engine as a function of engine load (expressed as indicated mean effective pressure, imep) for the three compression ratios and three engine speeds tested. It may be seen that both propane fumigation and liquid propane injection significantly reduce the smoke emissions, particularly at higher loads. Smoke emission of the liquid propane injection engine are slightly higher than those of the propane fumigation engine at 19:1 compression ratio but decrease to equivalent levels as the compression ratio is increased. Thus, for a given smoke level, an engine with a 25:1 compression ratio employing propane fumigation or liquid propane injection can produce approximately 15% more power than the original diesel.

This is an important consideration as emission standards for heavy duty vehicles are about to be implemented.

The tests with liquid propane injection revealsed that the relative position of the glow plug injection revealed that the relative position of the glow plug and the fuel spray was critical to obtaining good ignition and shor t ignition delays. The final arrangement used provided excellent results througout the engine's operating range at 25:1 compression ratio. However, at the two lower compression ratios, it was not possible to obtain reliable ignition at loads below about two-thirds of full load. Another arrangement of glow plug and injector provide ignition at light loads with the 19:1 comrpession ratio but had excessive rates of pressure rise at loads exceeding approximately 50% of full load. It is believed that reliable ignition can be attained under all engine operating conditions with some additional development.

Figures A.10 through A.18 show the percentage of total fuel energy provided by the diesel fuel ("diesel injection rate") as a function of engine load for the propane fumigated engine. With fumigation, the amount of diesel fuel required was sensitive to the compression ratio, increasing as the compression ratio increased. This is because the limit to the amount of propane fumigation that could be used was the onset of knock as measured by the rate of pressure rise. Knock occurs more readily at higher compression ratios. From a fuel substitution standpoint, liquid propane injection with glow plug ignition is more desirable than fumigation because it permits complete substitution of propane for diesel fuel.

Indicated specific fuel consumption is hsown in Figures A.19 through A.27 as a function of load for the three fueling methods. The energy consumption of the liquid propane injection engine is equal to that of the diesel at 25:1 compression ratio especially at high loads and high speeds. However, at lower compression ratios, the energy consumption of the liquid propane injection engine is higher than tha to the diesel.

The same trend is maintained for fumigation. At lower compression ratios, the energy consumption falls with load until the fuel consumption of the propane fumigated engine is equivalent to that of the original diesel at full load. At higher compression ratios, the energy consumption is not very sensitive to load and is significantly better than that of the original diesel. At 25:1 compression ratio, the fuel load indicated specific fuel consumptions were 11 MJ/kW-h, 9 MJ/kW-h, and 8 MJ/kW-h for diesel, liquid propane inejction and propane fumigation respectively. Thus, the fuel consumption is improved 18% with liquid propane injection and 27% with propane fumigation.

Plots of the hydrocarbon emissions (Figures A.28 to A.36) and CO emissions (Figures A.37 to A.45) show evidence of incomplete combustion in the fumigated engine at light loads. The emissions of the fumigated engine are high at light loads and drop steadily as full load operation is reached. Increasing the compression ratio appears to improve combustion at light loads as the light load emissions are reduced at the higher compression ratios. The liquid propane injection engine has emission levels comparable to those of the diesel, except at light loads and high compression ratio where there is a slight increase. The overall levels remain very low however.

Engine load has completely opposite effects on the NO_x emissions of the fumigated engine and the liquid propane injection engine (Figures A.46 to A.54). In general, the NO_x emission of the liquid propane injeciton engine are lower than those of the original diesel at light loads and decrease more as load is increased. With fumigation, the NO_x emissions are very low at light loads but increase sharply as full load is approached, especially at low compression ratios. For both propane fueling systems, the influence of load was reduced by increased compression ratio.

5. CONCLUSIONS

In summary, the use of propane fuel in diesel engines offers the significant advantage of reduced smoke emissions while achieving a power otuput equivalent to the original diesel. Of the two propane fueling methods, liquid propane injection with glow plug ignition offers complete propane substitution for diesel fuel, while as much as 25% of the total fuel energy must be supplied by diesel fuel in the case of fumigation. In addition, the liquid propane injection engine has lower NO_x emissions except at light loads. At the present state of development, the energy consumption of the liquid propane injection engine is equal to that of the diesel at 25:1 compression ratio, increasing somewhat at lower compression ratios. Fumigation provides excellent energy consumptio at full load operation. Where operation at light loads is a major part of the duty cycle, is not the preferred method of fuel delivery.

The tradeoff for the advantages of liquid propane injection is the increased mechanical complexity of the system. Considerable development is needed to provide a simpler and more reliable engine fueling system. A number of problem areas were identified in this first stage of development which should be addressed in subsequent work.

The first of the problems encountered is keepign the propane fuel liquified. Heat input to saturated liquid propane creates cavitation problems in the supply pump that prpagate through the fuel injection pump and cause erratic injector operation. In the vehicle application, a propane pump must be installed near the tank to increase the propane pressure before the propane encounters high underhood temperatures. This need not be a pressure above the critical pressure if heat conduction from the engine block to the fuel injection pump can be minimized. A high pressure line would be needed to convey the fuel to the engine compartment. Natural gas fueled vehicles already have a much higher pressure fuel supply line running the length of the vehicle.

Wear problems in the propane fuel injection pump are still not completely solved. Use of alternative materials for wear surfaces may be necessary. Perhaps a coating, e.g. Teflon, would provide the necessary protection. The flow of lubricating oil applied to the pump barrel as in this project this is not well ocntrolled and could result in significant oil consumption. Furthermore, lubricating oil appears to provide ignition even in the absence of the diesel pilot, but with high smoke emissions. Resonance in the fuel injeciton line was also a problem and resulted in late secondary injections of propane and subsequent high smoke emissions. This is not a problem peculiar to propane, but requires careful tailoring of the injection system to the particular installation.

These problems should be overcome with further development. The results of the first stage of development carried out in this project demonstrate the feasibility and promise of the liquid propane injection diesel engine with glow plug ignition.

6. REFERENCES

1. UTME/ERDL Reports for Ontario Ministry of Transportation and Communications. "Propane Fuel/Engine Systems Optimization and Testing", Phases I, II, III, IIIb, IV and V, 1981-83.

2. Karim, G.A. "A Review of Combustion Processes in the Dual Fuel Engine - The Gas Diesel Engine", Prog. Energy Combust. Sci., 6, pp. 277-285, 1980.

3. Pischinger, F. "Dieselzweistoffbetrieb mit Flüssigeinspritzung von Propane/Butane ins Ansaugrohr", MTZ Motortech A., 40 (3), pp. 121-124, 1979.

4. Oester, U., Wallace, J.S., Segal, L., Keffer, J.F. "Propane-Fueled Diesel Engines", UTME/ERDL Report for SNG Inc., 1986.

APPENDIX

This appendix contains the experimental results in graphic form.

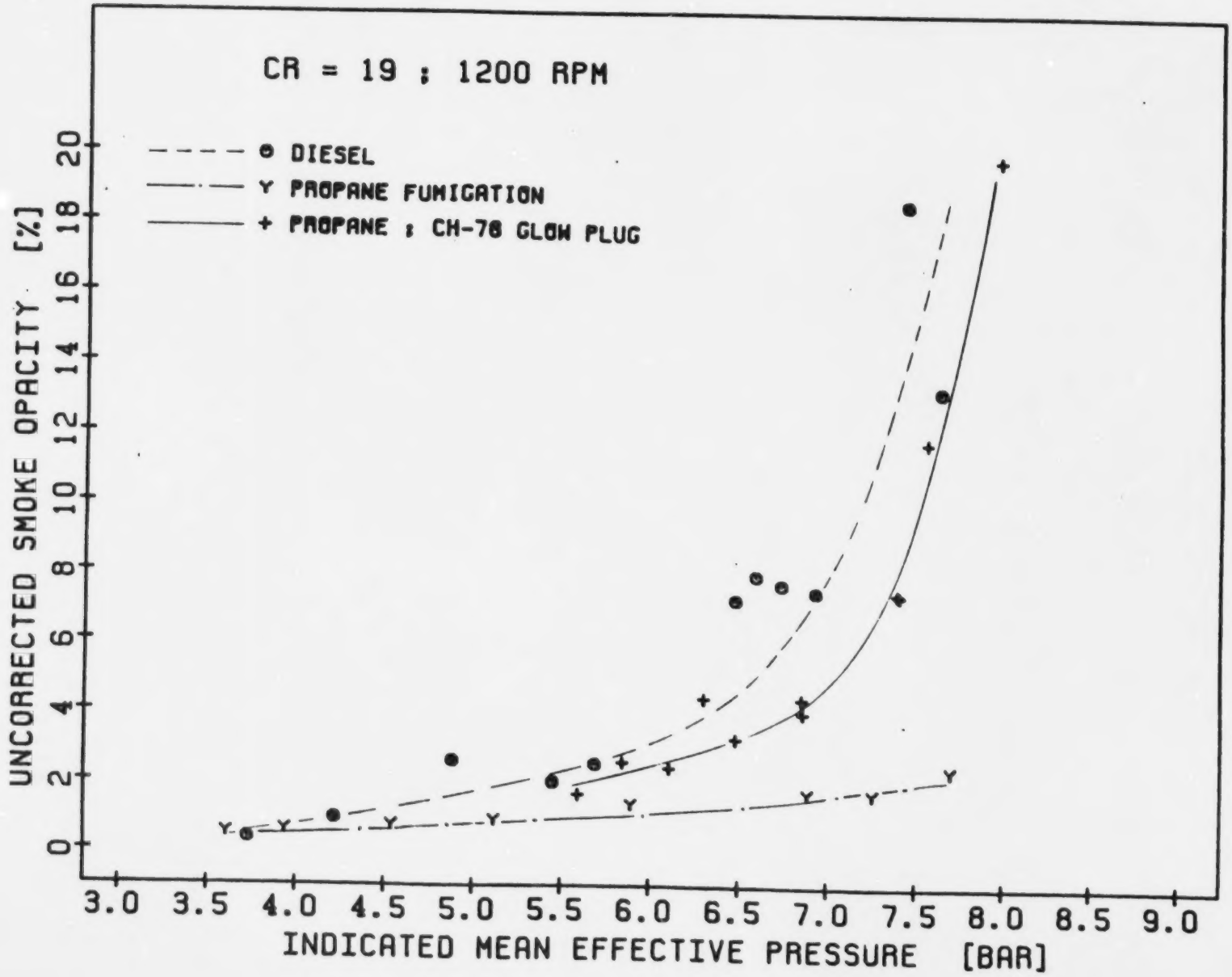


Figure A.1 Uncorrected smoke opacity vs. load as IMEP (CR = 19; 1200 RPM)

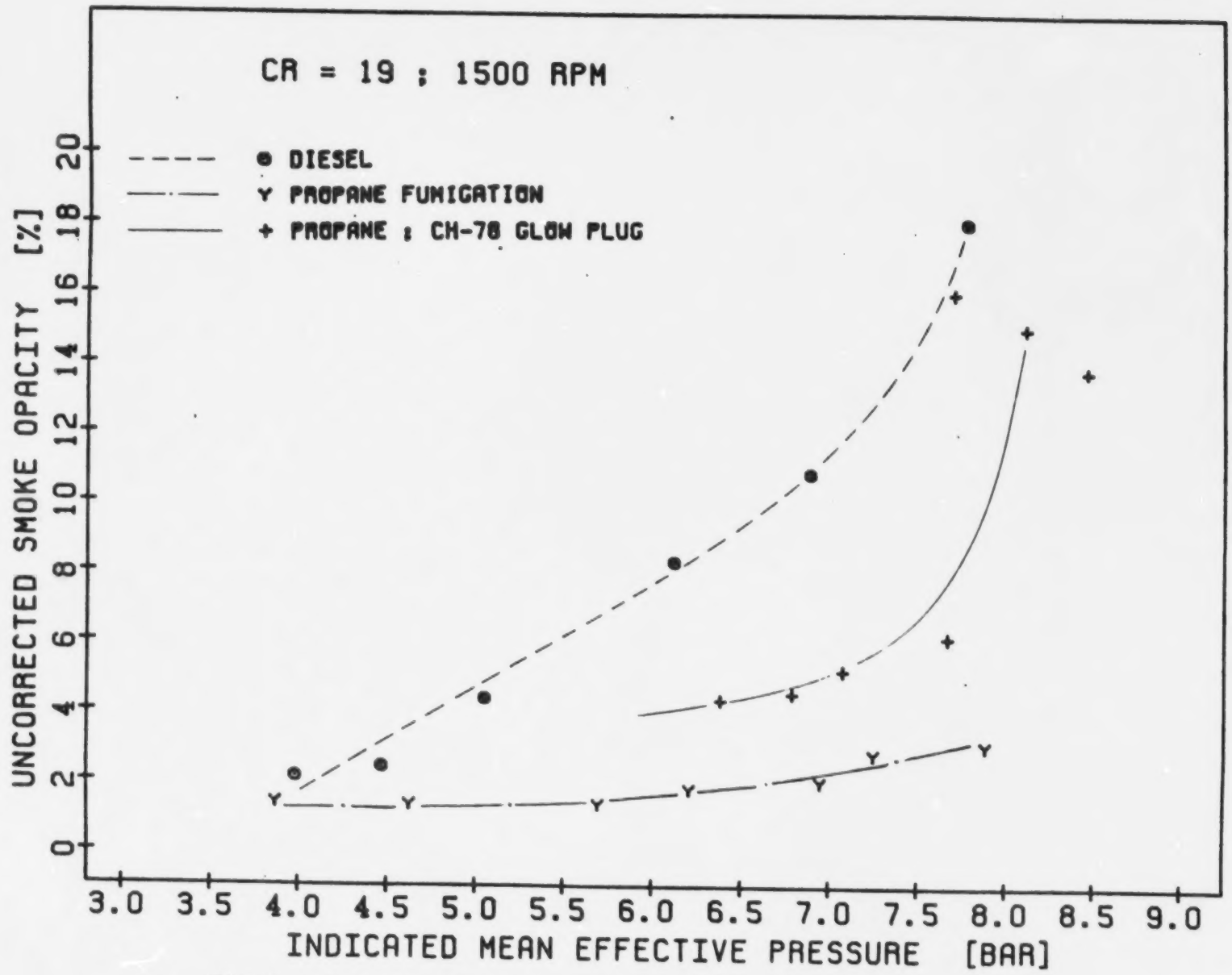


Figure A.2 Uncorrected smoke opacity vs. load as IMEP (CR = 19; 1500 RPM)

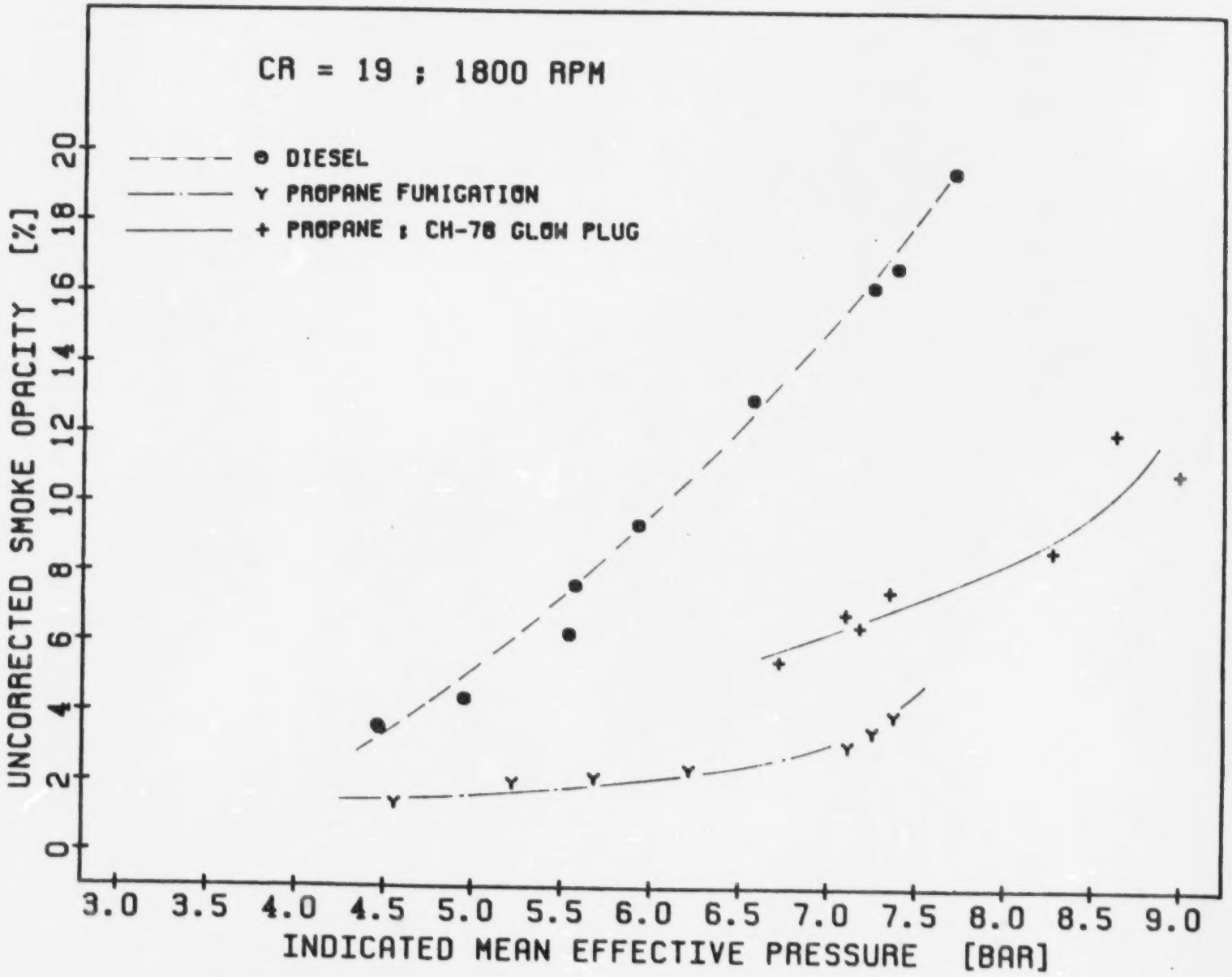


Figure A.3 Uncorrected smoke opacity vs. load as IMEP (CR = 19; 1800 RPM)

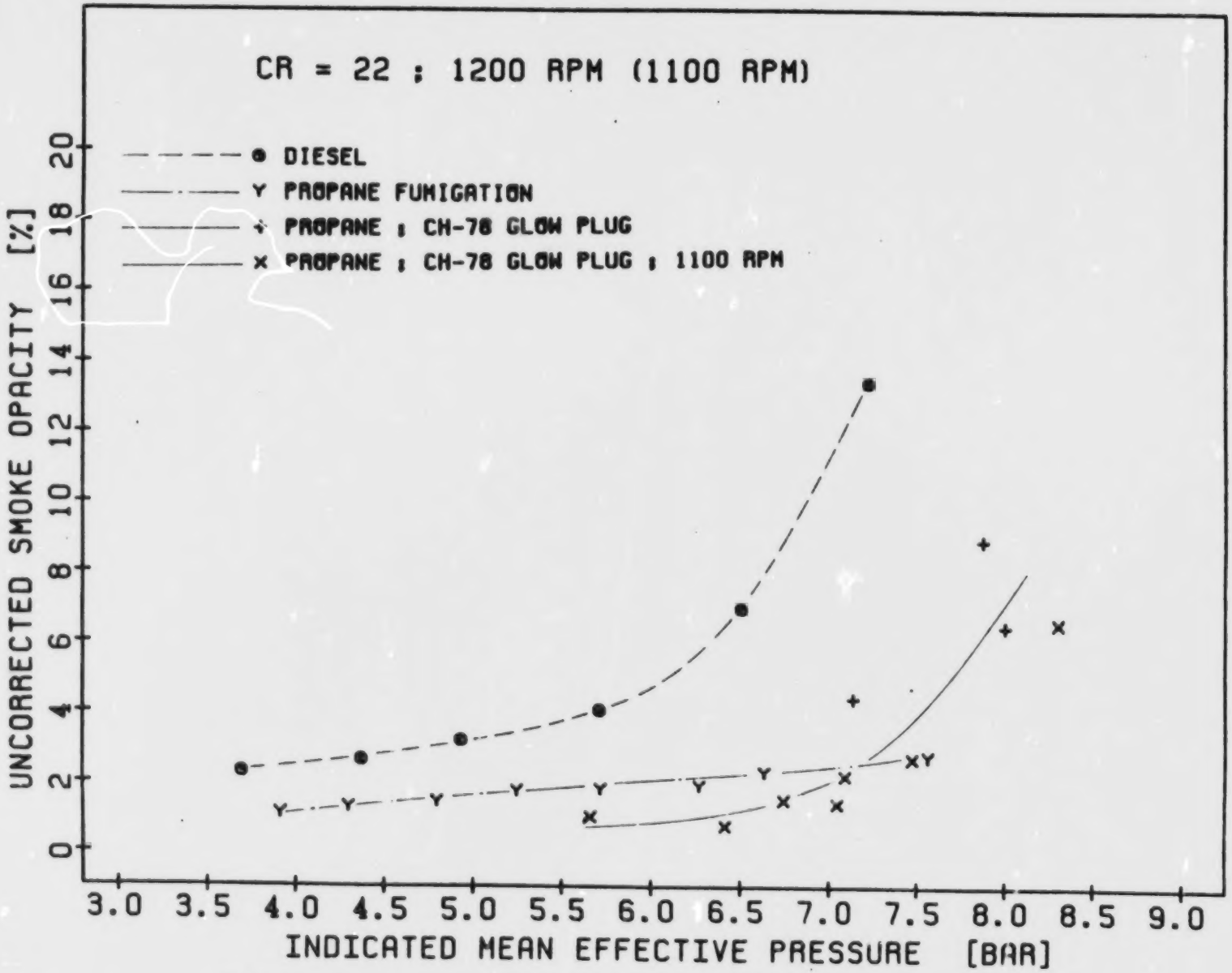


Figure A.4 Uncorrected smoke opacity vs. load as IMEP (CR = 22; 1200 RPM)

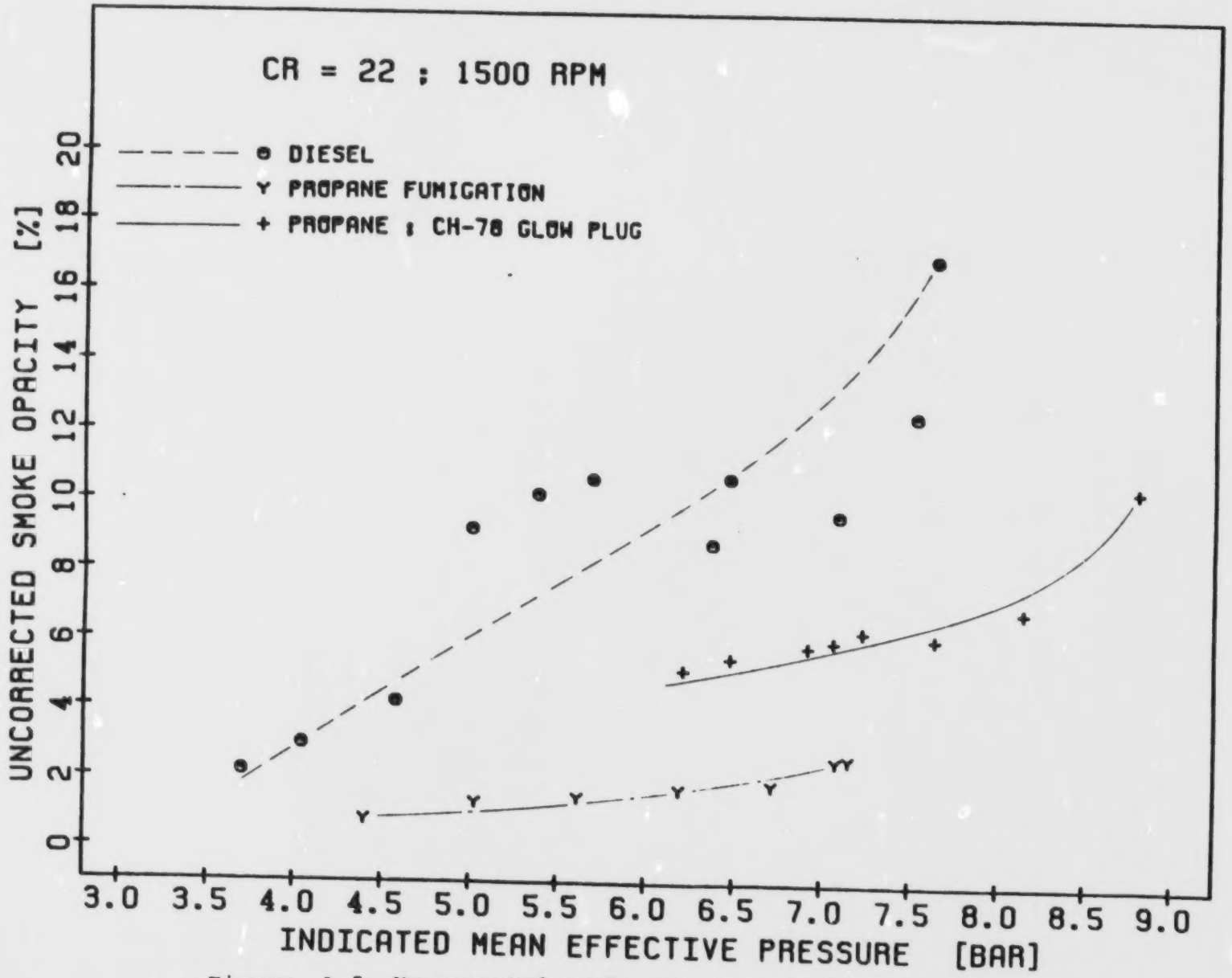


Figure A.5 Uncorrected smoke opacity vs. load as IMEP (CR = 22; 1500 RPM)

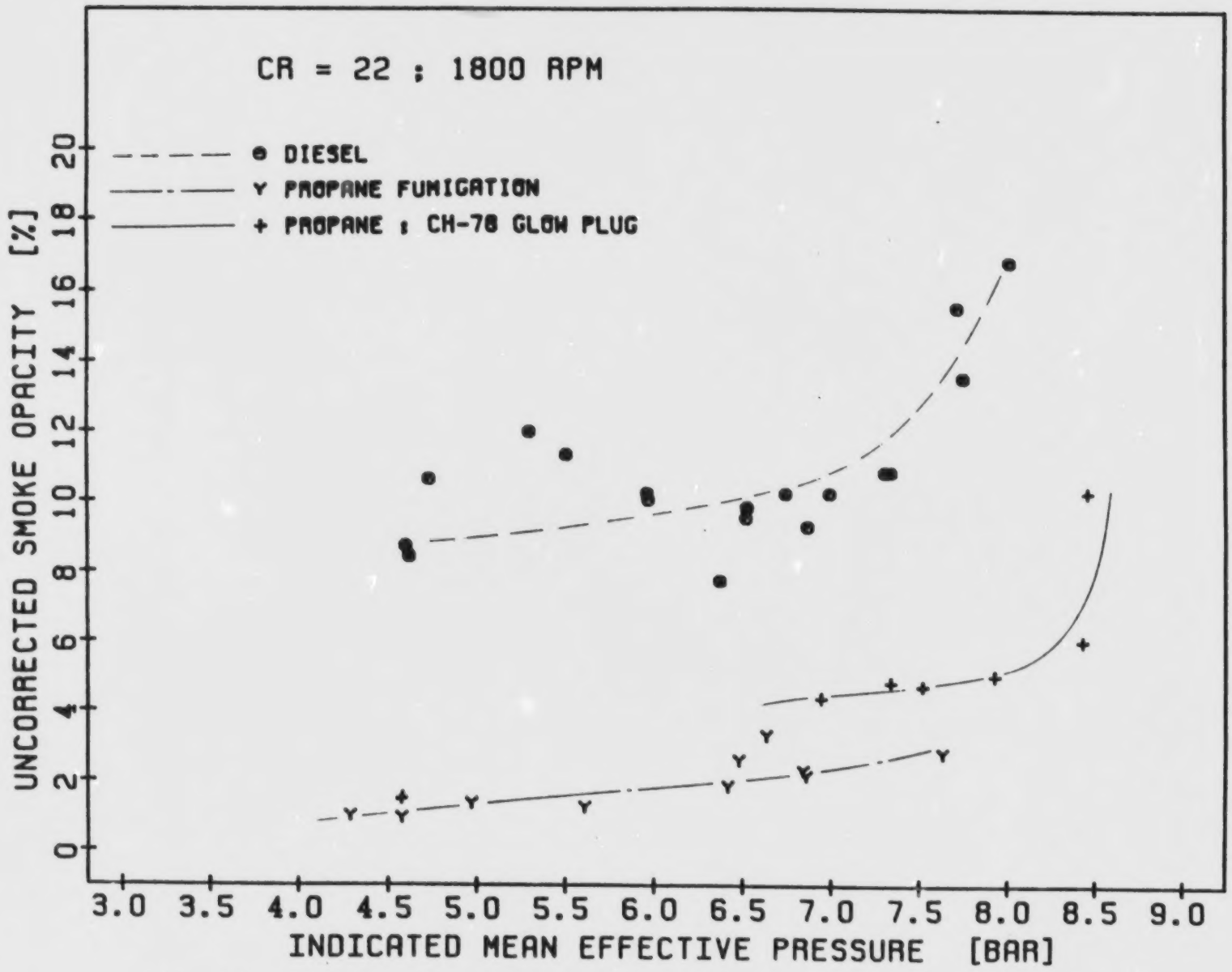


Figure A.6 Uncorrected smoke opacity vs. load as IMEP (CR = 22; 1800 RPM)

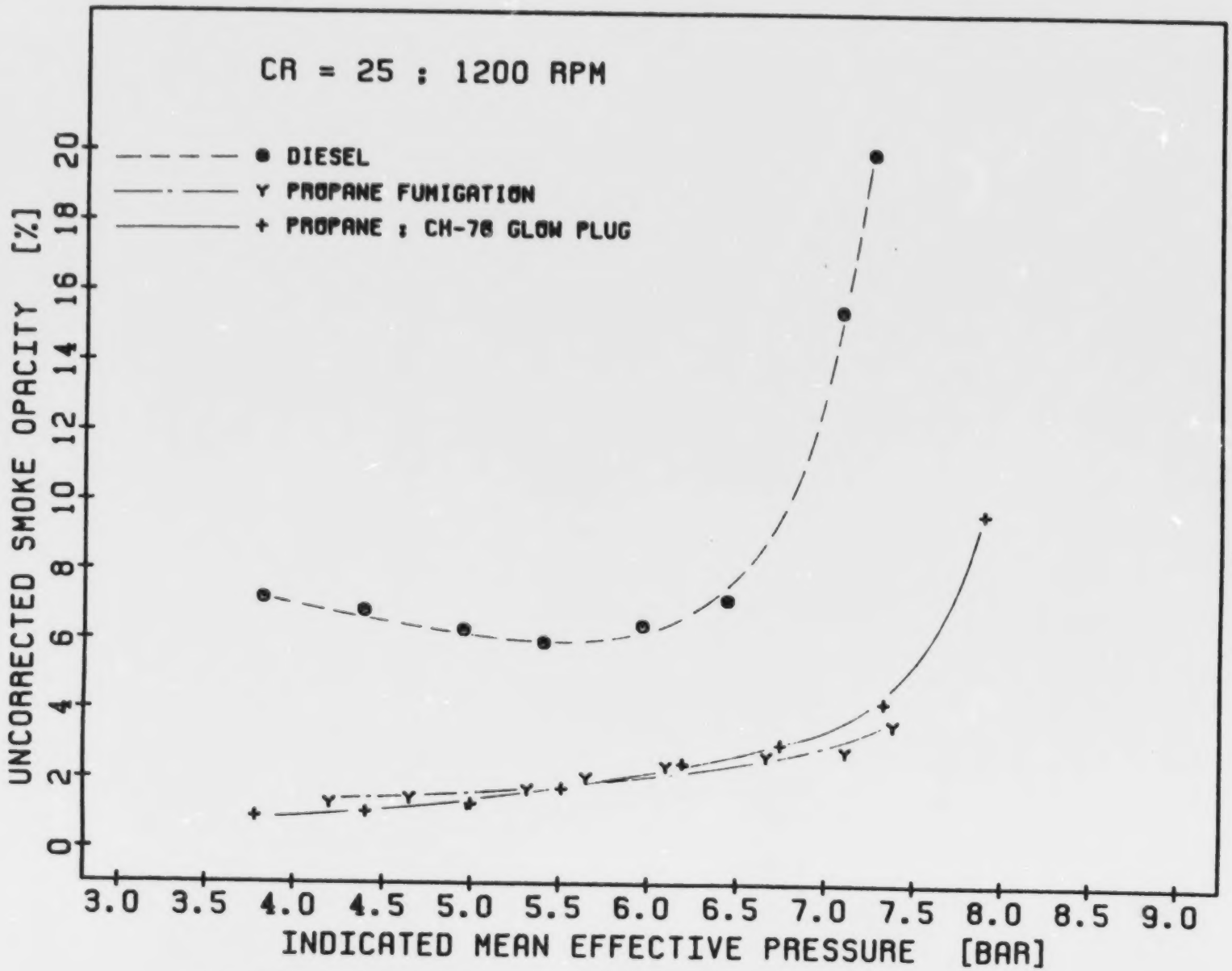


Figure A.7 Uncorrected smoke opacity vs. load as IMEP (CR = 25; 1200 RPM)

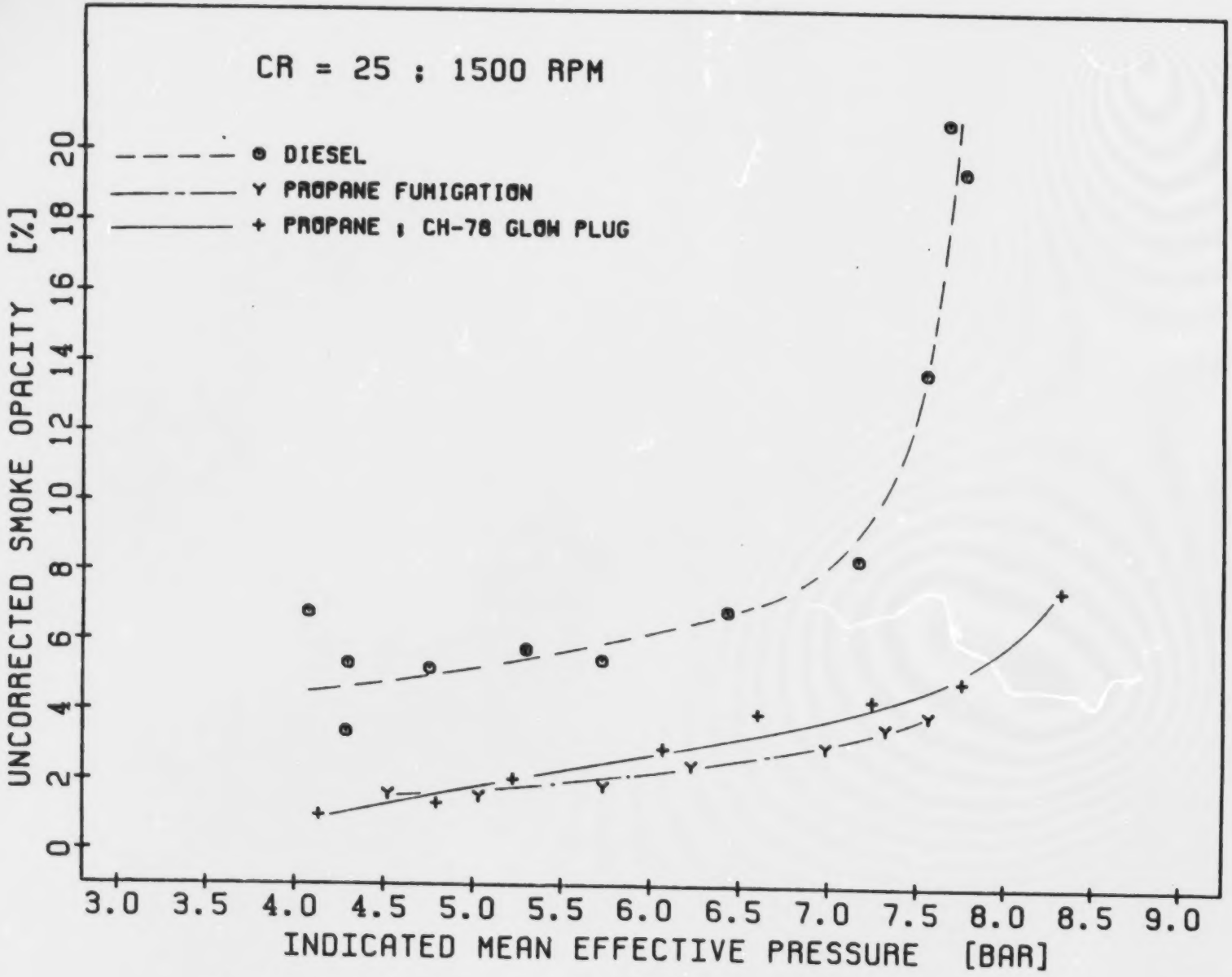


Figure A.8 Uncorrected smoke opacity vs. load as IMEP (CR = 25; 1500 RPM)

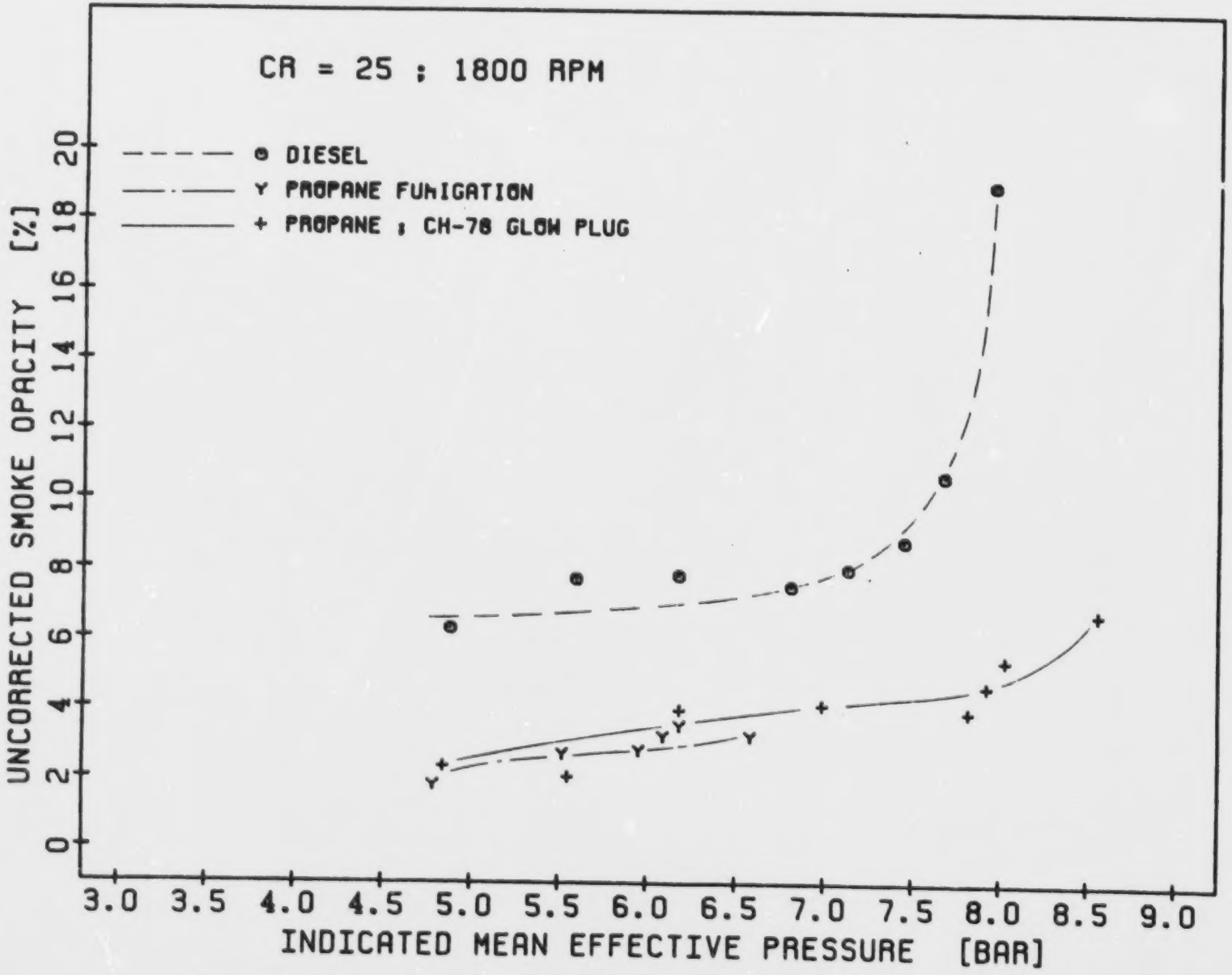


Figure A.9 Uncorrected smoke opacity vs. load as IMEP (CR = 25; 1800 RPM)

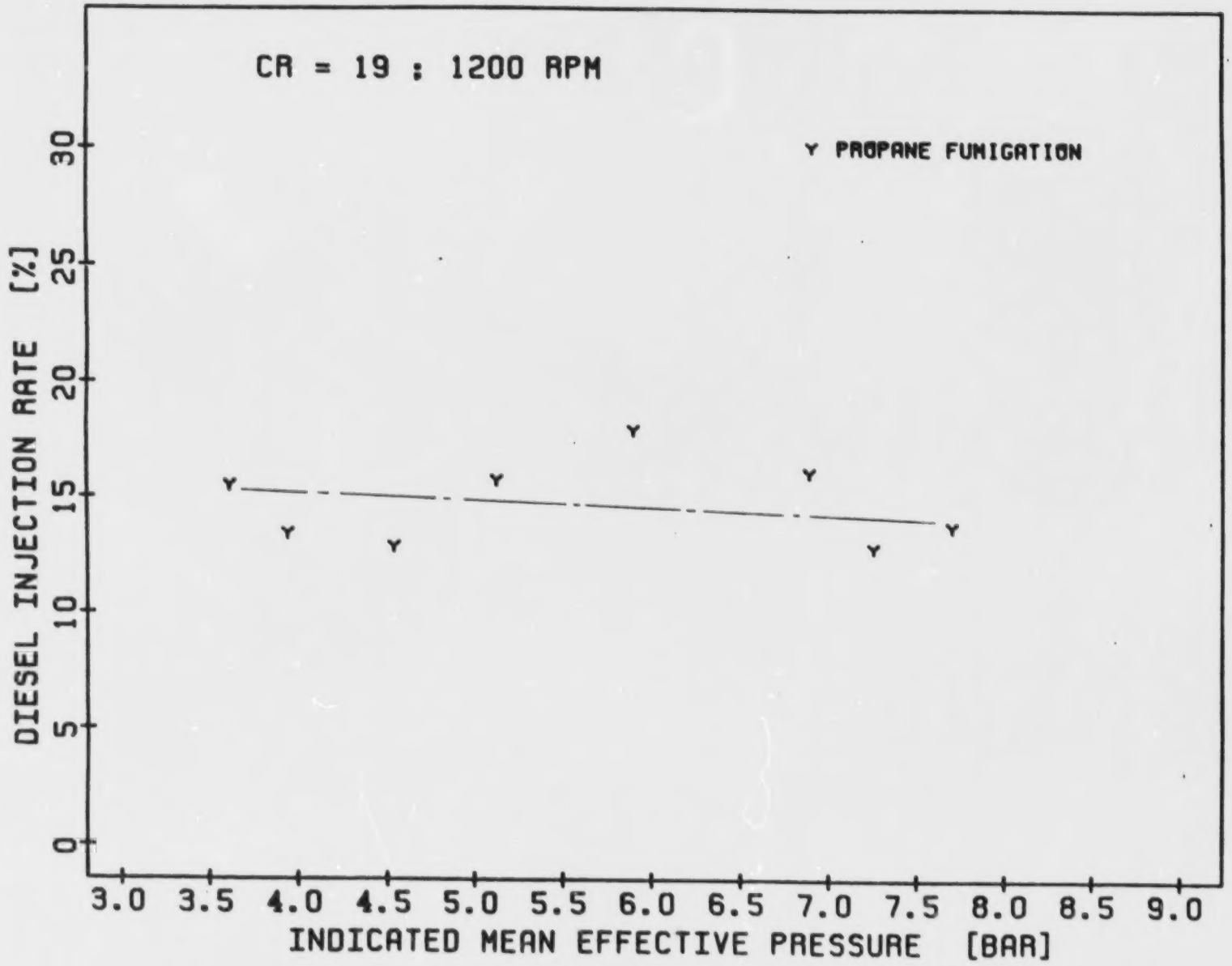


Figure A.10 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 19; 1200 RPM)

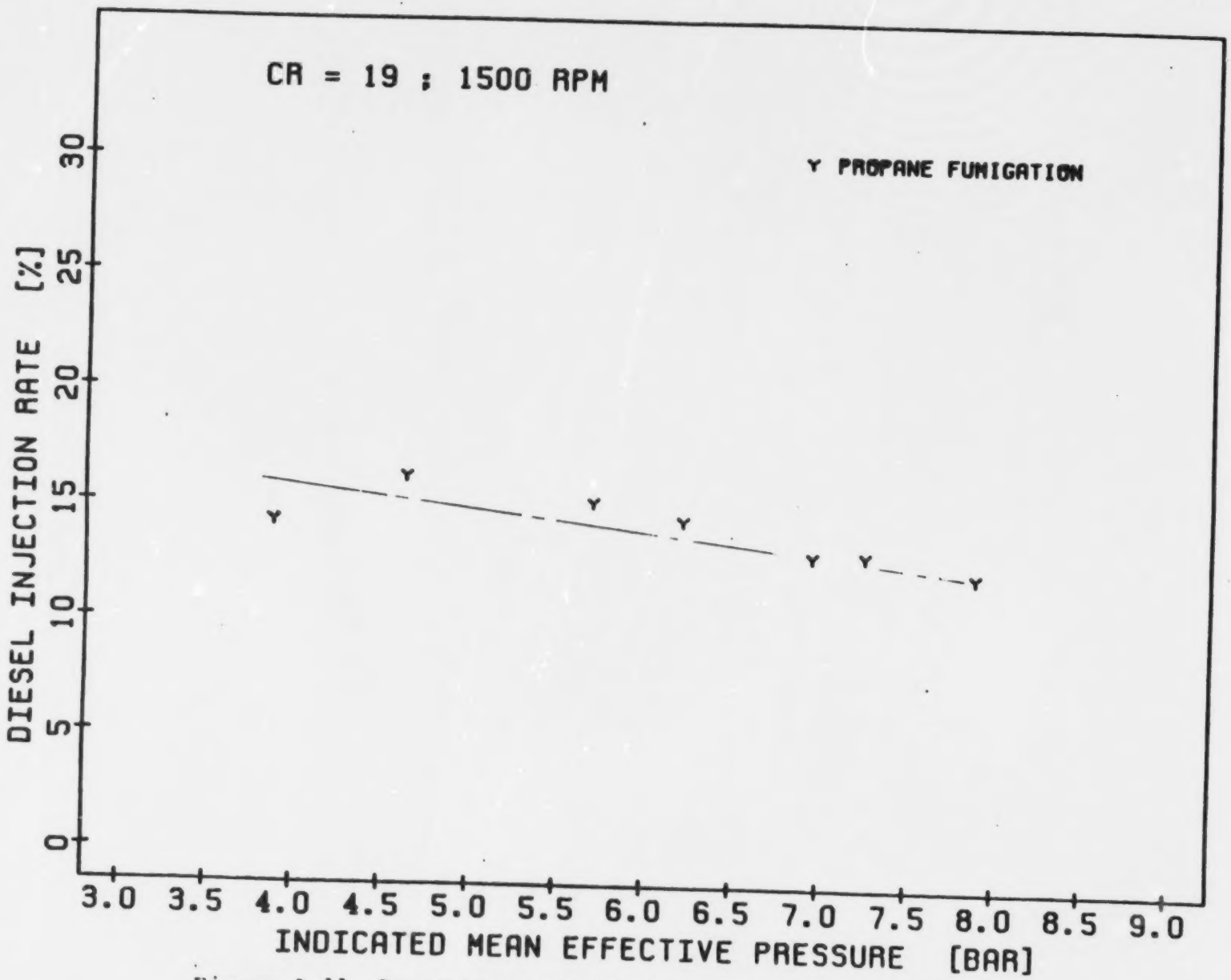


Figure A.11 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 19; 1500 RPM)

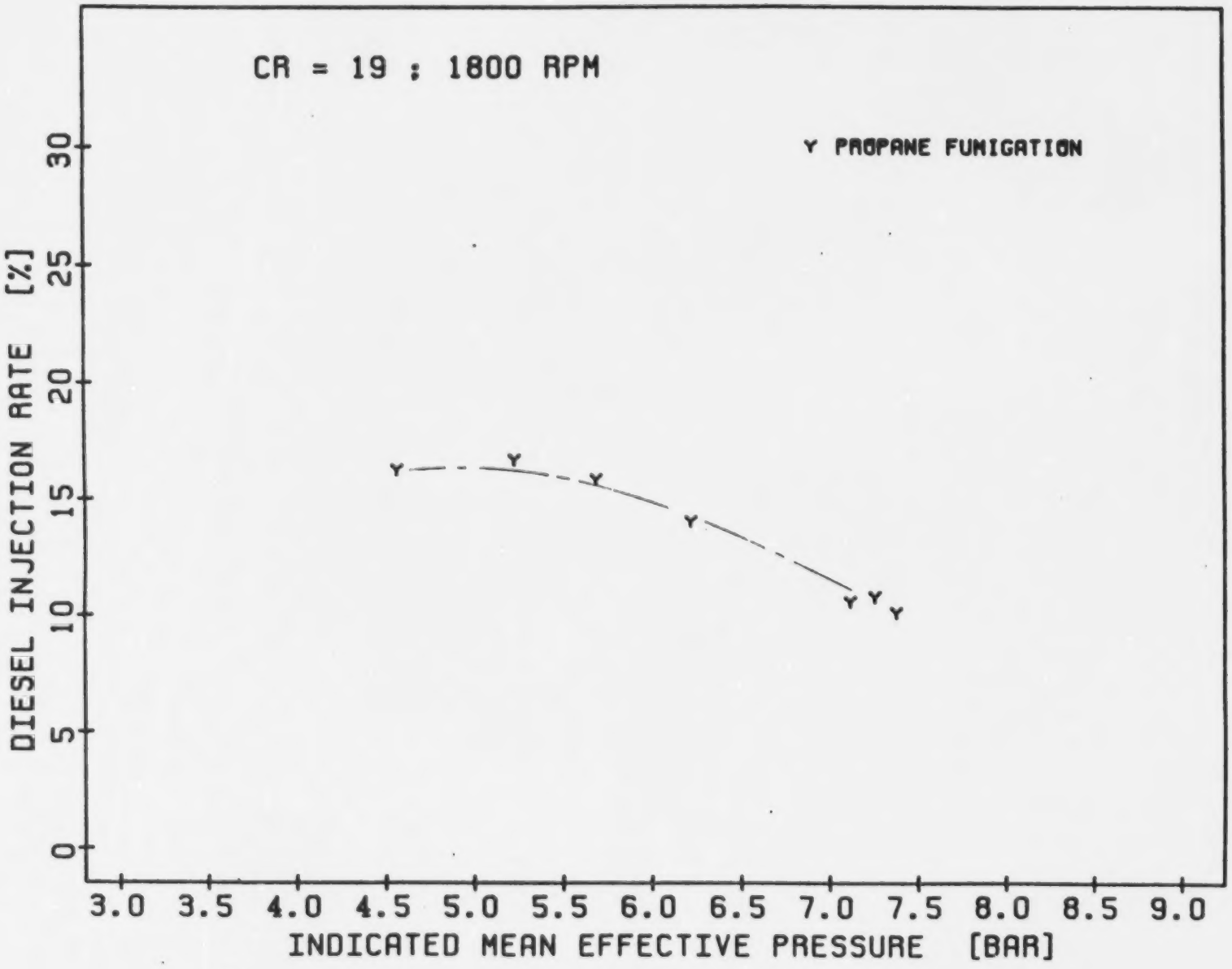


Figure A.12 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 19; 1800 RPM)

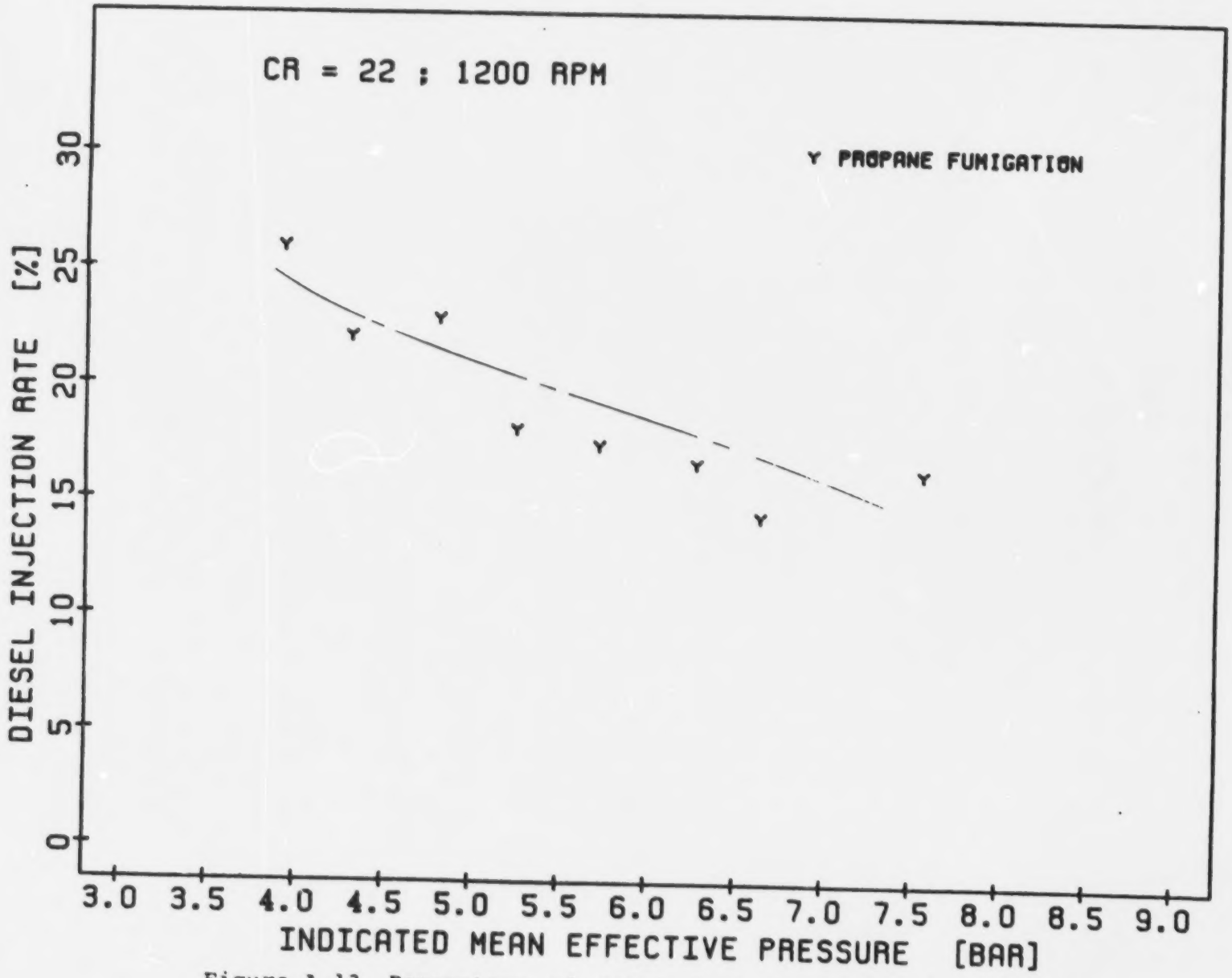


Figure A.13 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 22; 1200 RPM)

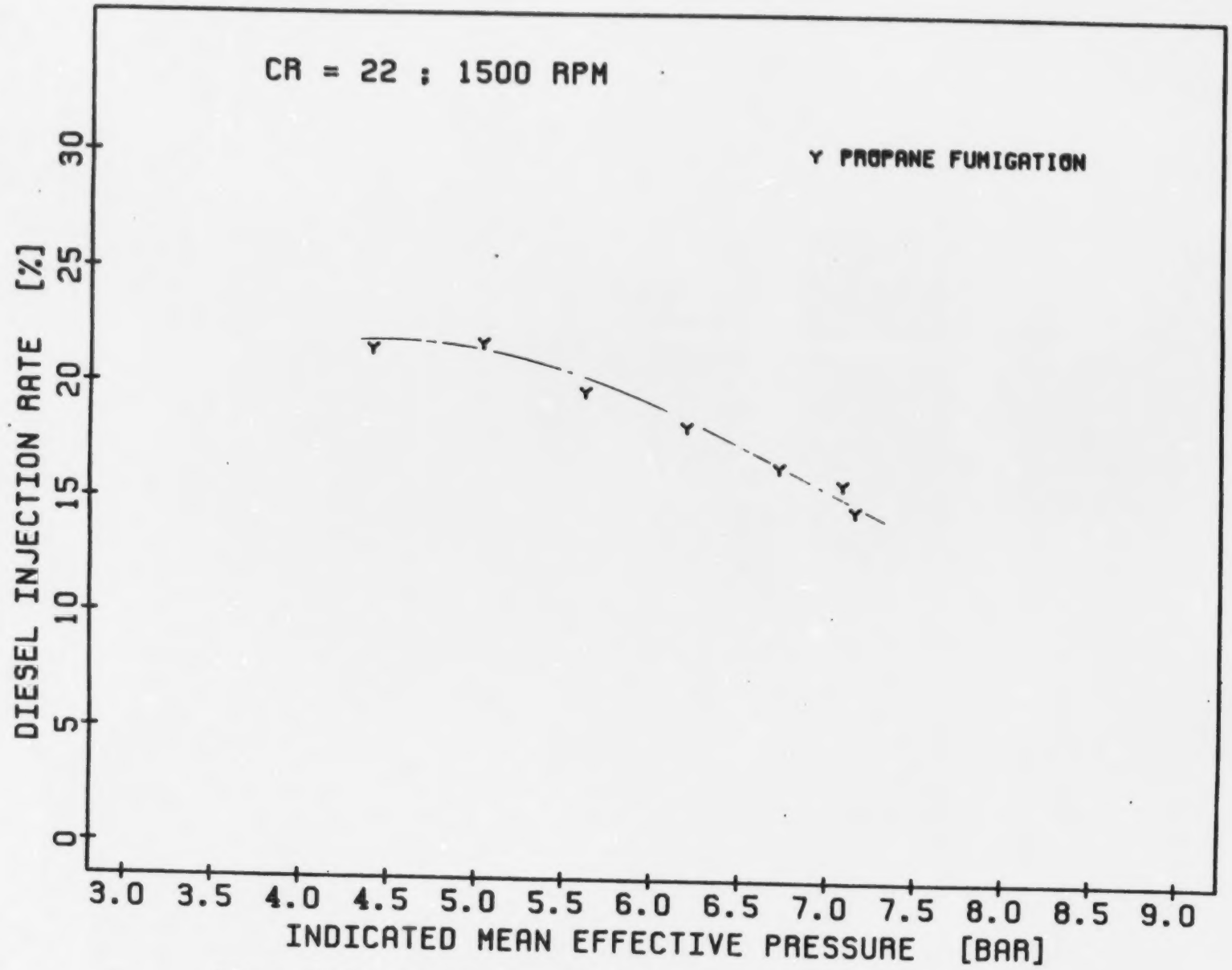


Figure A.14 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 22; 1500 RPM)

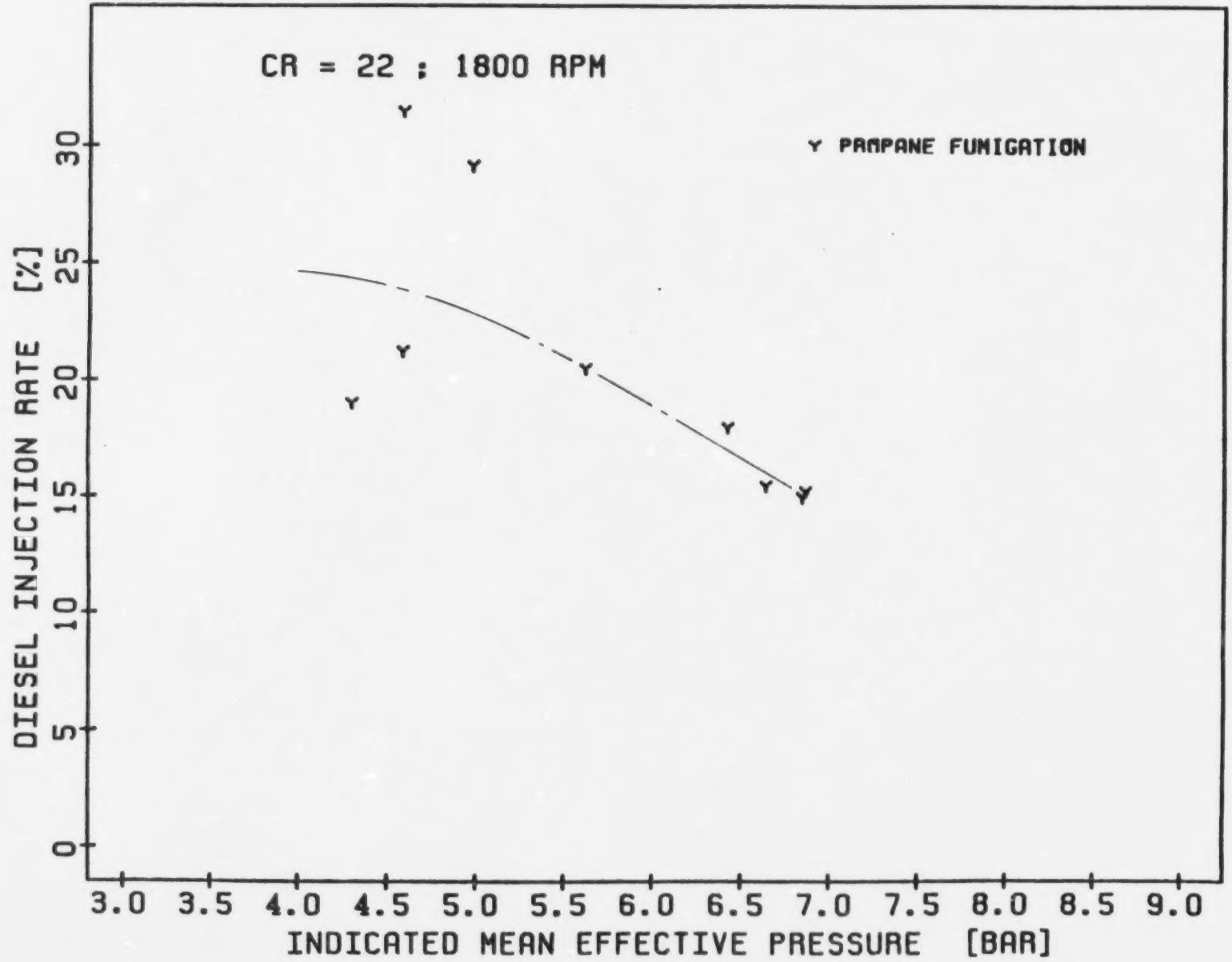


Figure A.15 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 22; 1800 RPM)

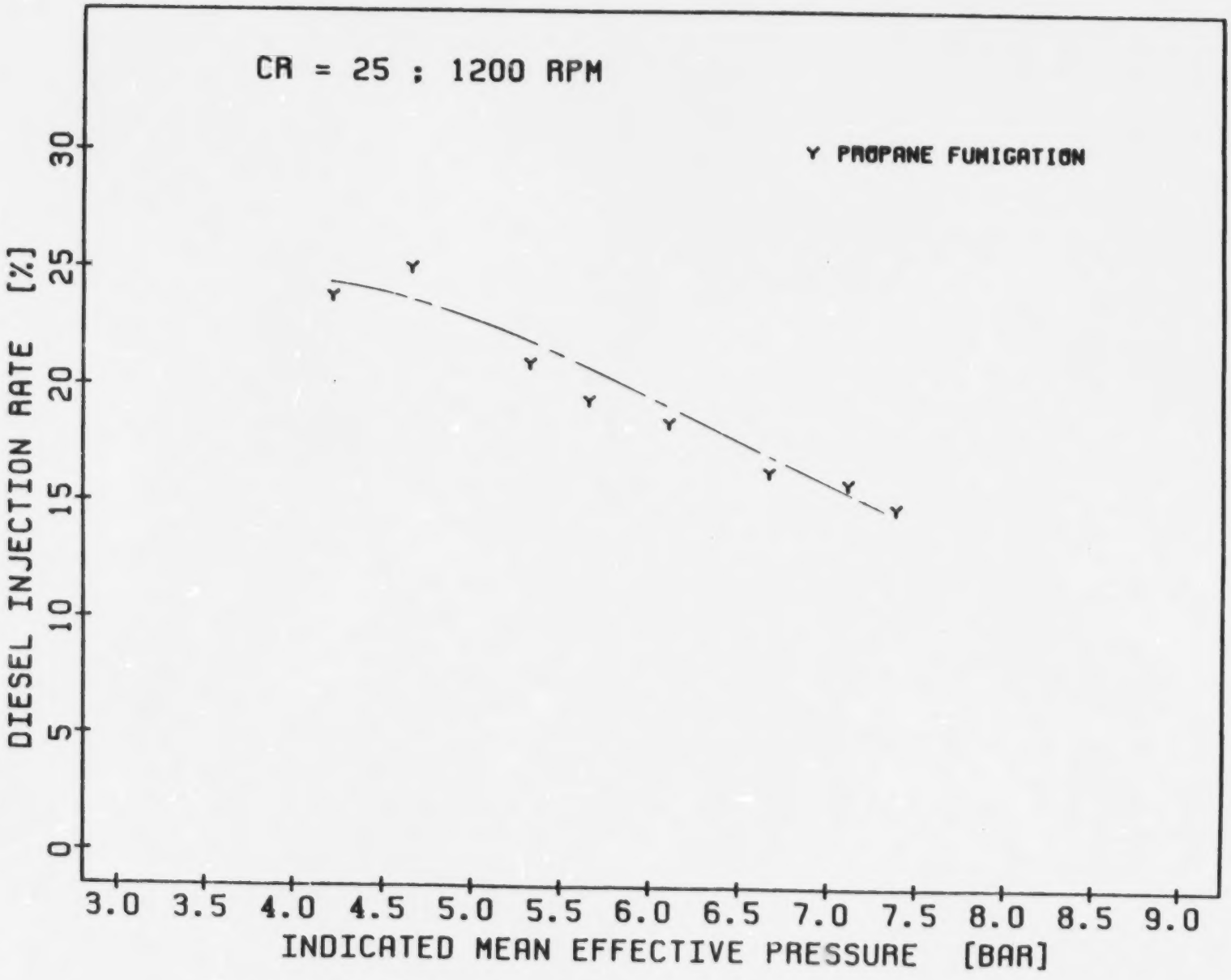


Figure A.16 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 25, 1200 RPM)

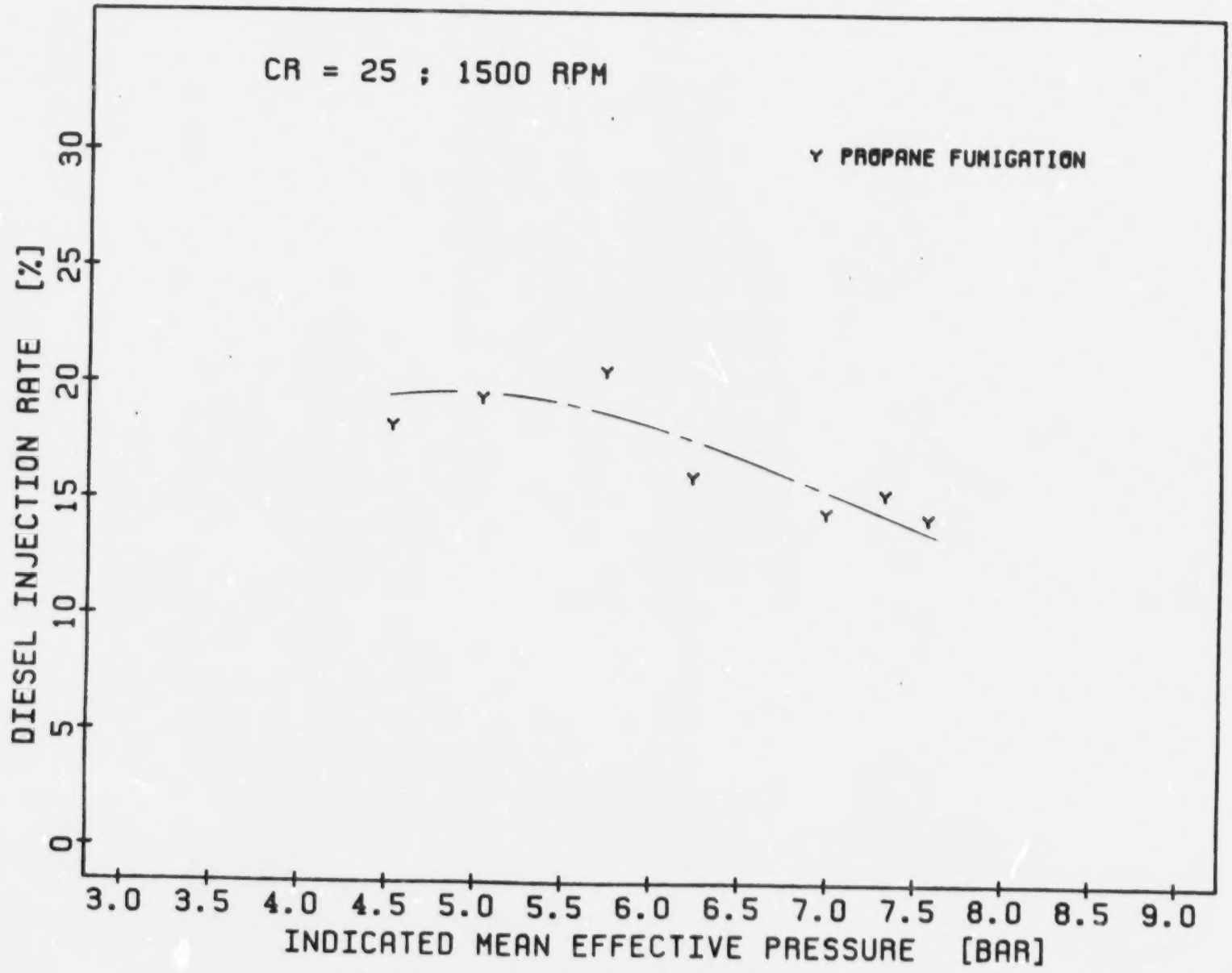


Figure A.17 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 25; 1500 RPM)

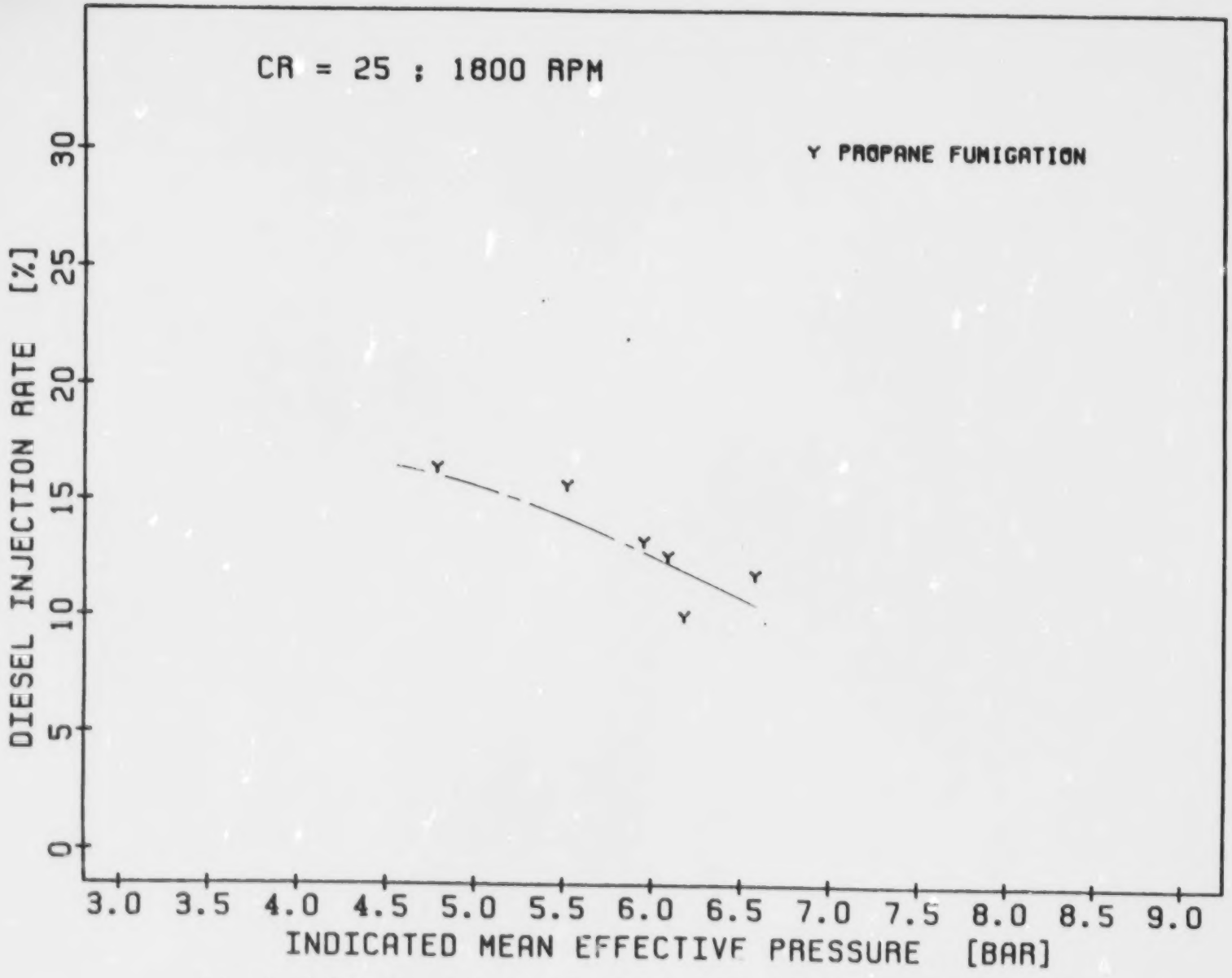


Figure A.18 Percentage of total fuel energy provided by the diesel fuel vs. load as IMEP (CR = 25; 1800 RPM)

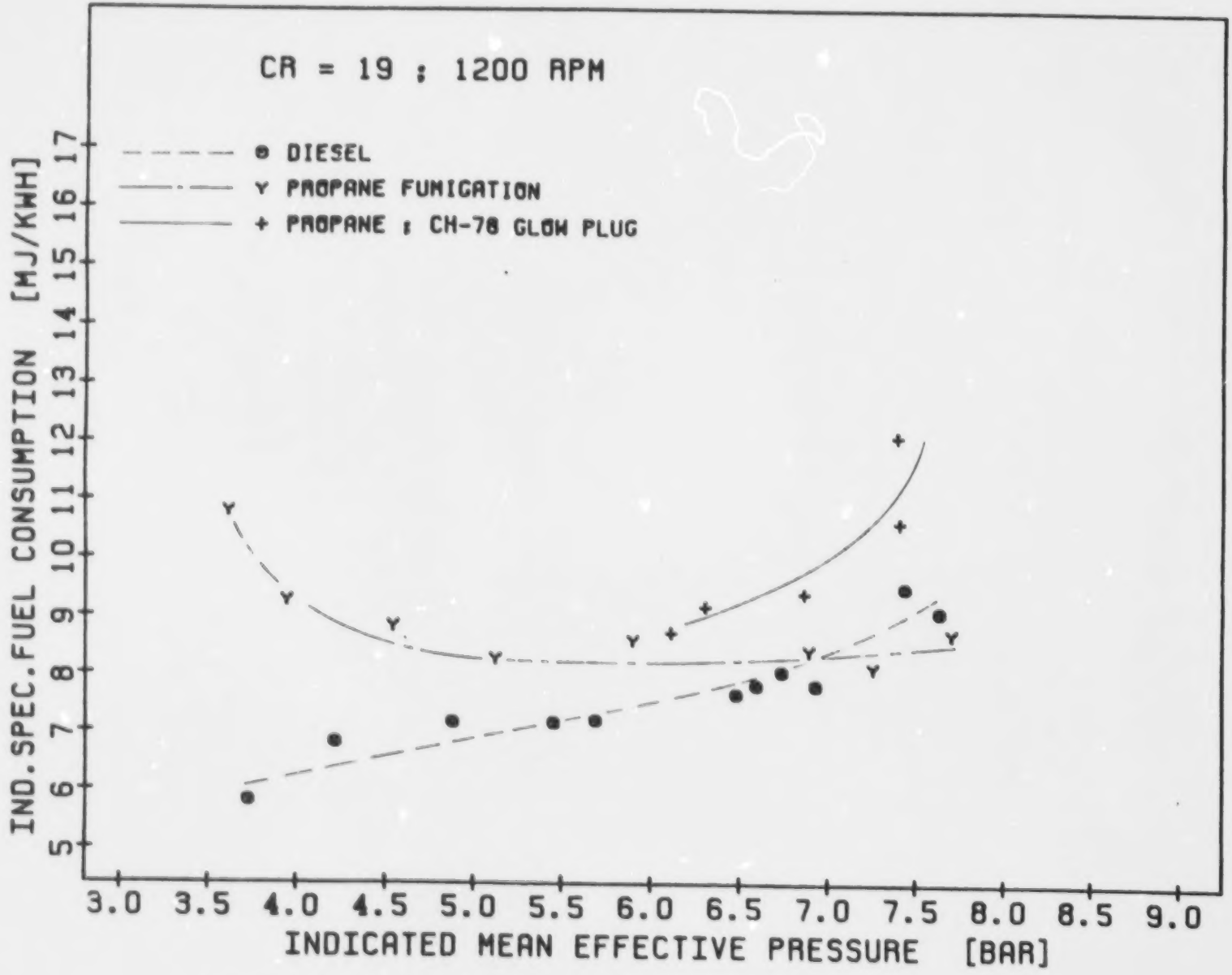

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.19 Indicated specific fuel consumption vs. load as IMEP (CR = 19; 1200 RPM)

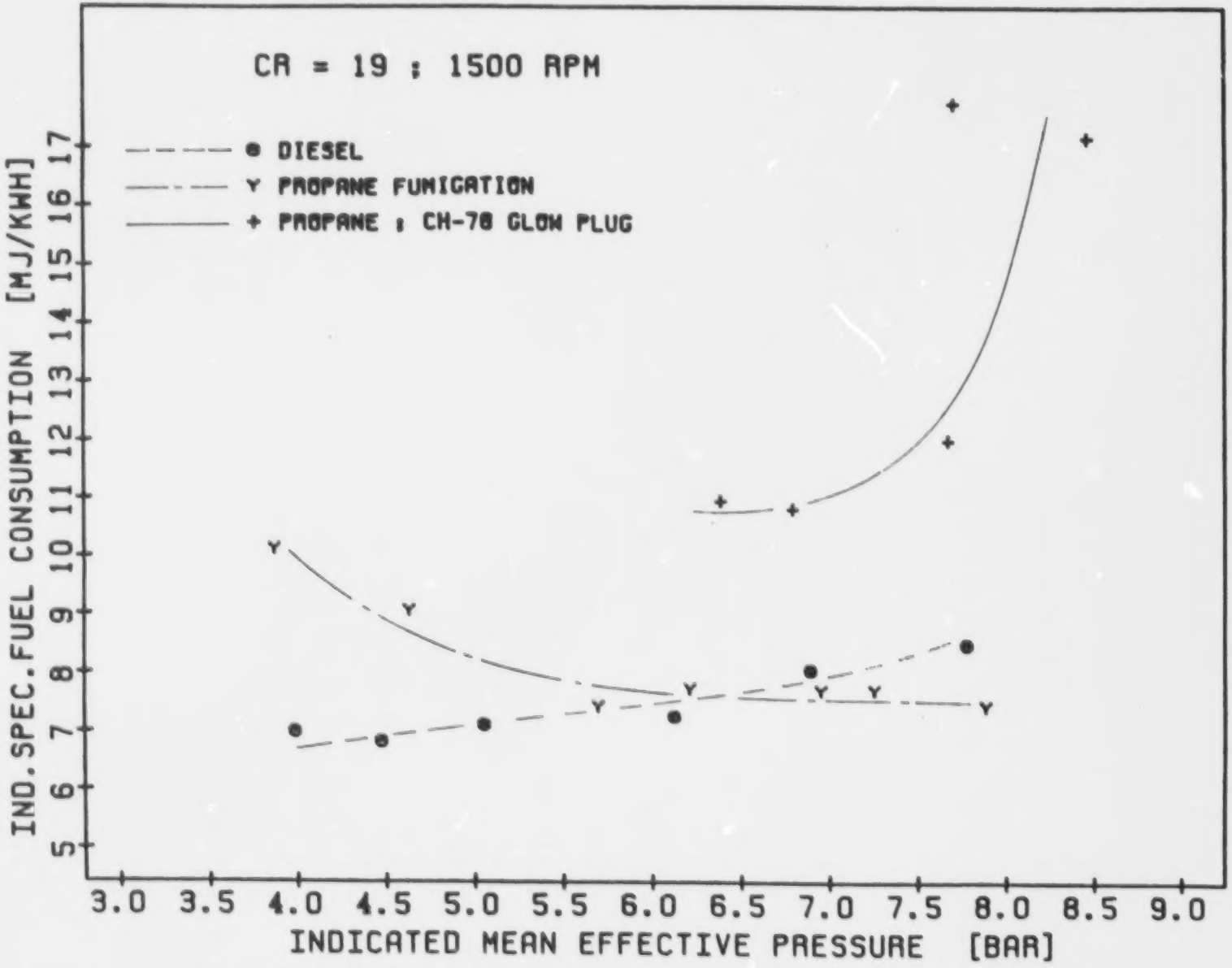

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.20 Indicated specific fuel consumption vs. load as IMEP (CR = 19; 1500 RPM)

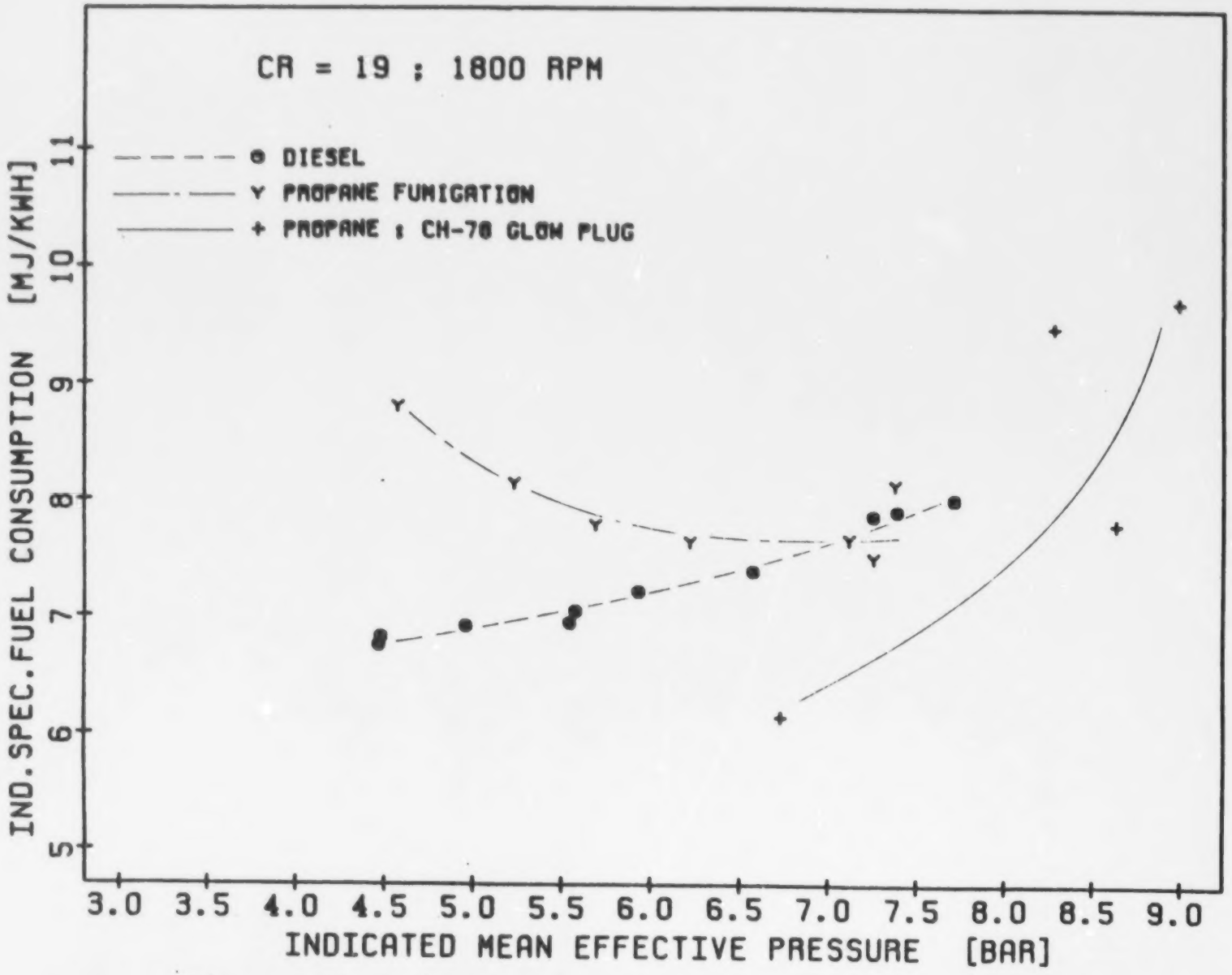


Figure A.21 Indicated specific fuel consumption vs. load as IMEP (CR = 19; 1800 RPM)

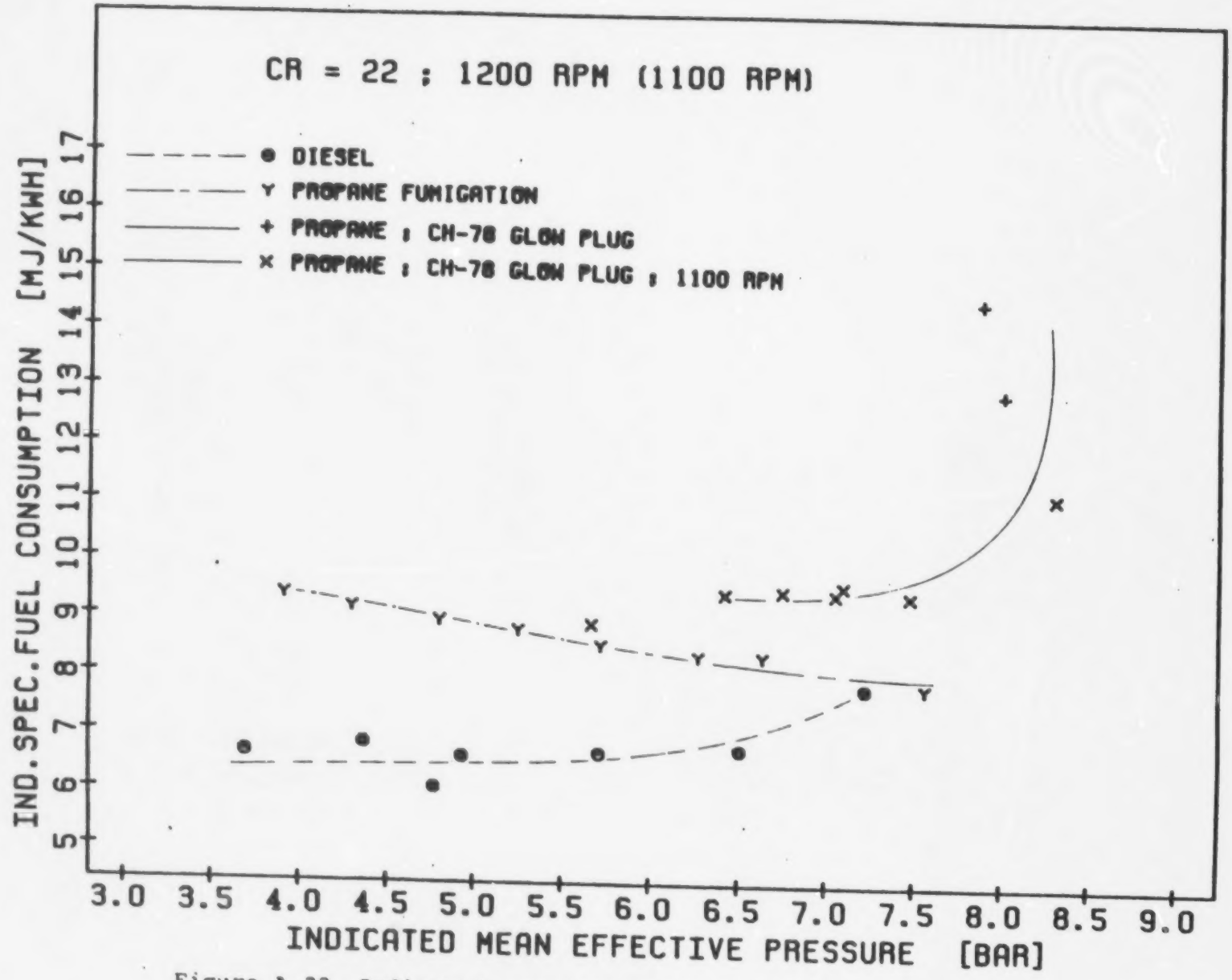

CR = 22 ; 1200 RPM (1100 RPM)
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
PROPANE ; CH-78 GLOW PLUG ; 1100 RPM
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.22 Indicated specific fuel consumption vs. load as IMEP (CR = 22; 1200 RPM)

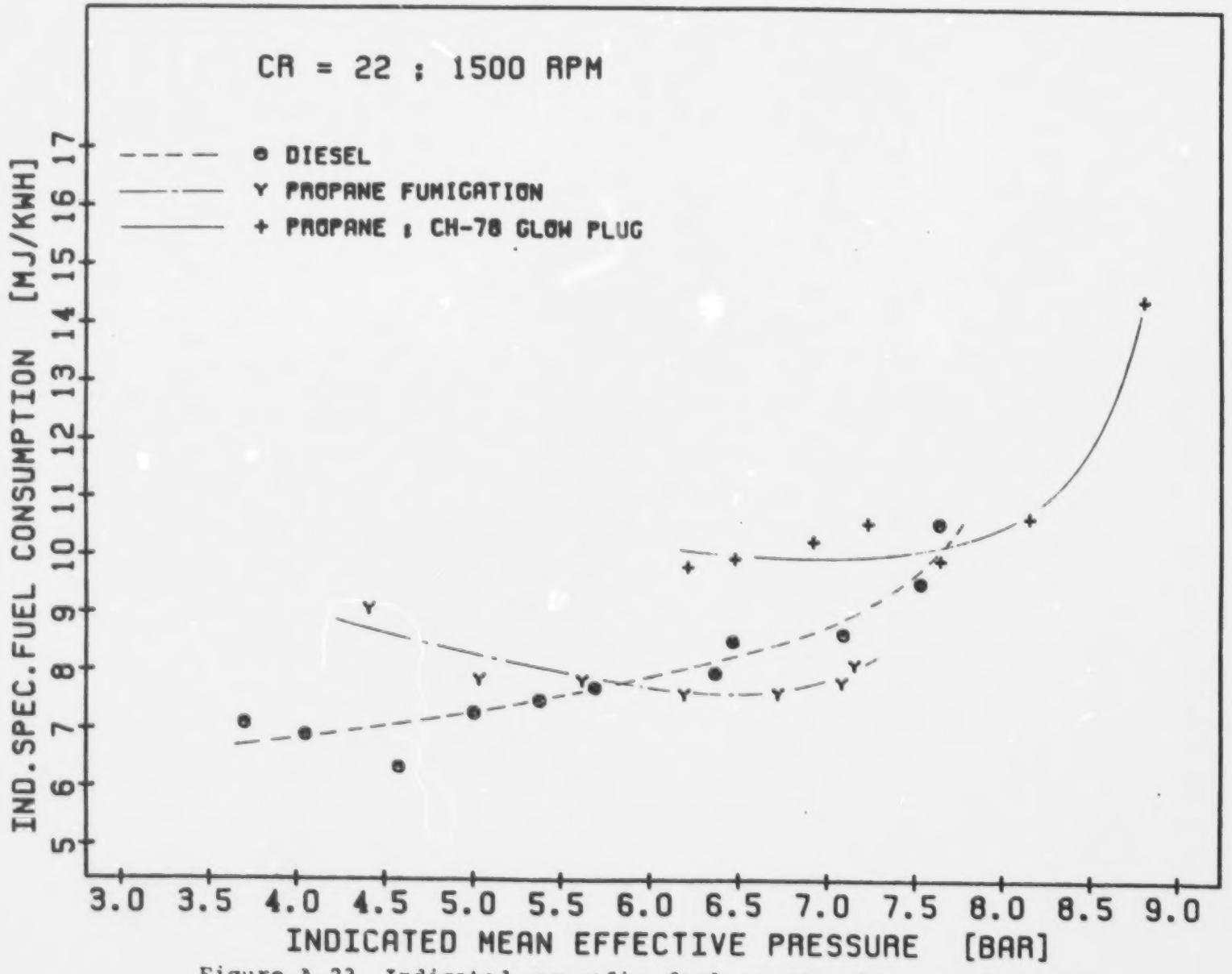

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.23 Indicated specific fuel consumption vs. load as IMEP (CR = 22; 1500 RPM)

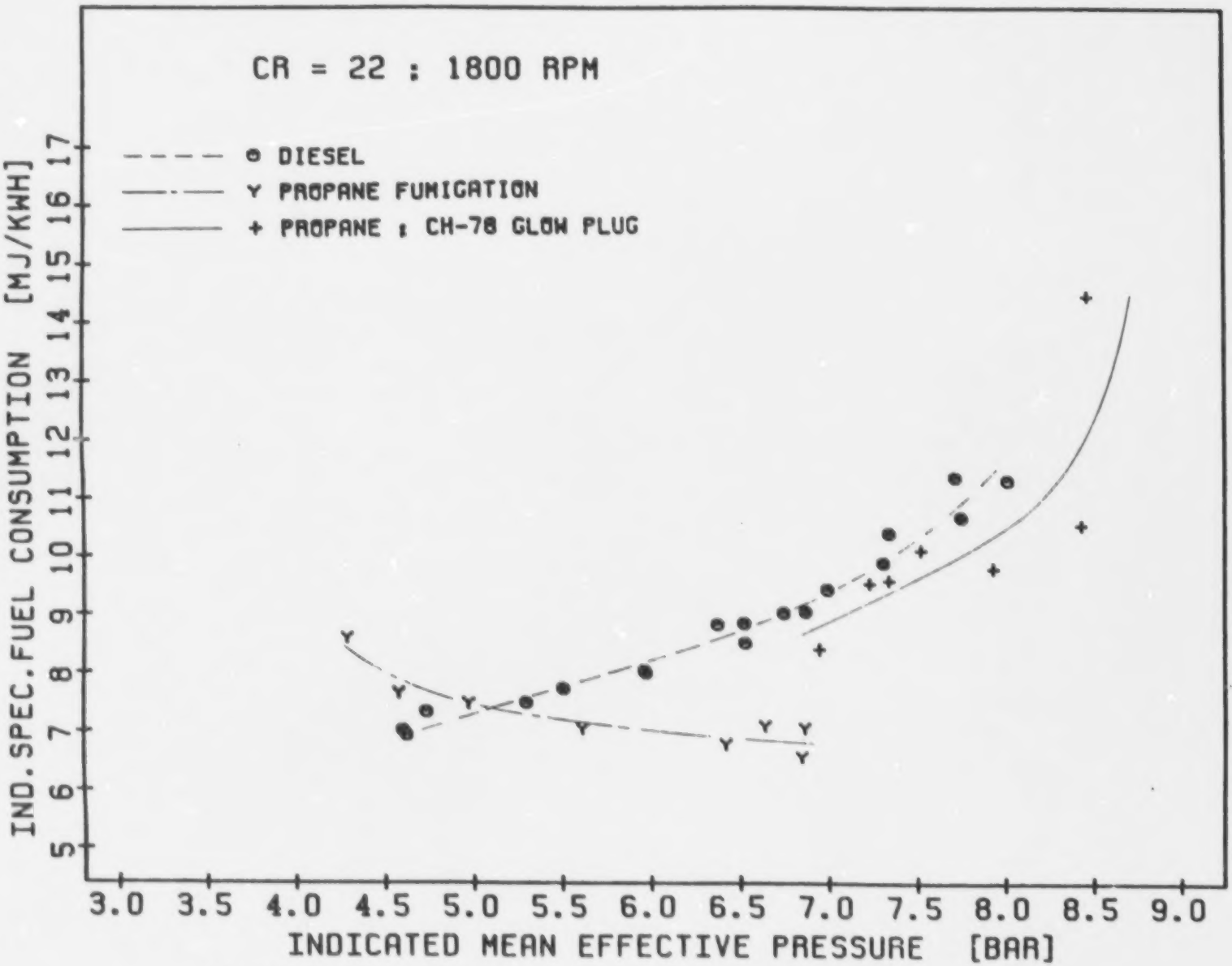

CR = 22 : 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE : CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.24 Indicated specific fuel consumption vs. load as IMEP (CR = 22; 1800 RPM)

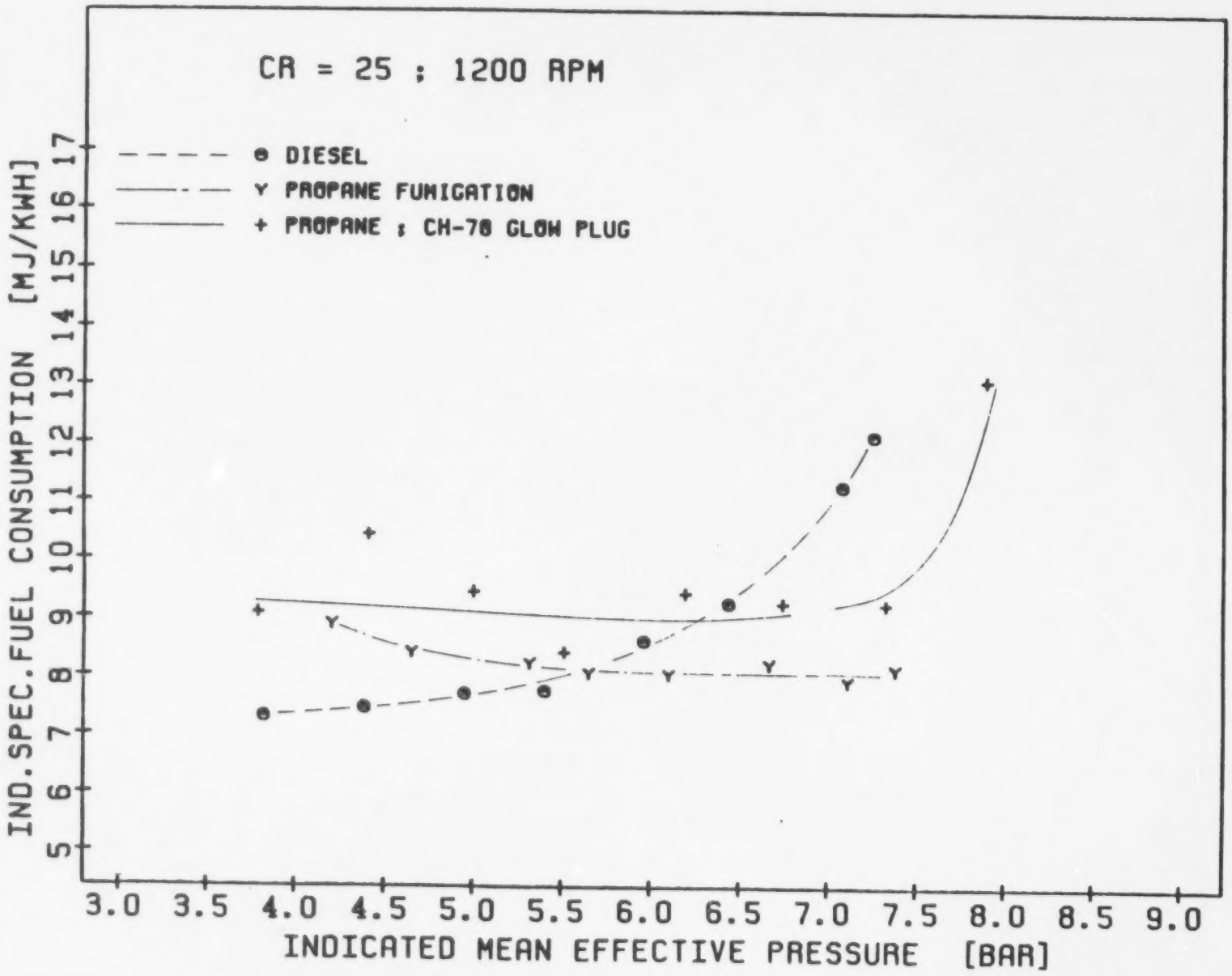

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.25 Indicated specific fuel consumption vs. load as IMEP (CR = 25; 1200 RPM)

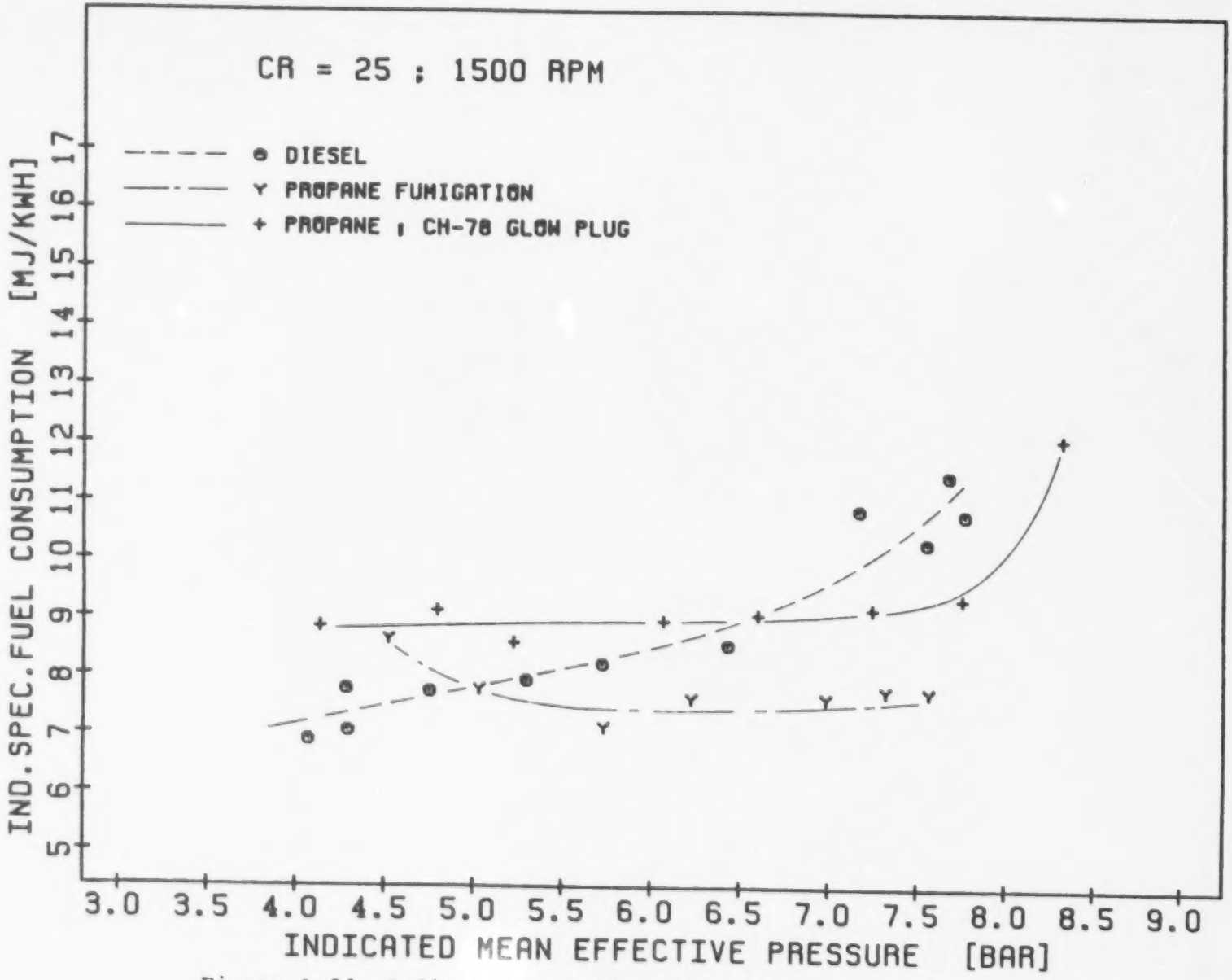

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.26 Indicated specific fuel consumption vs. load as IMEP (CR = 25; 1500 RPM)

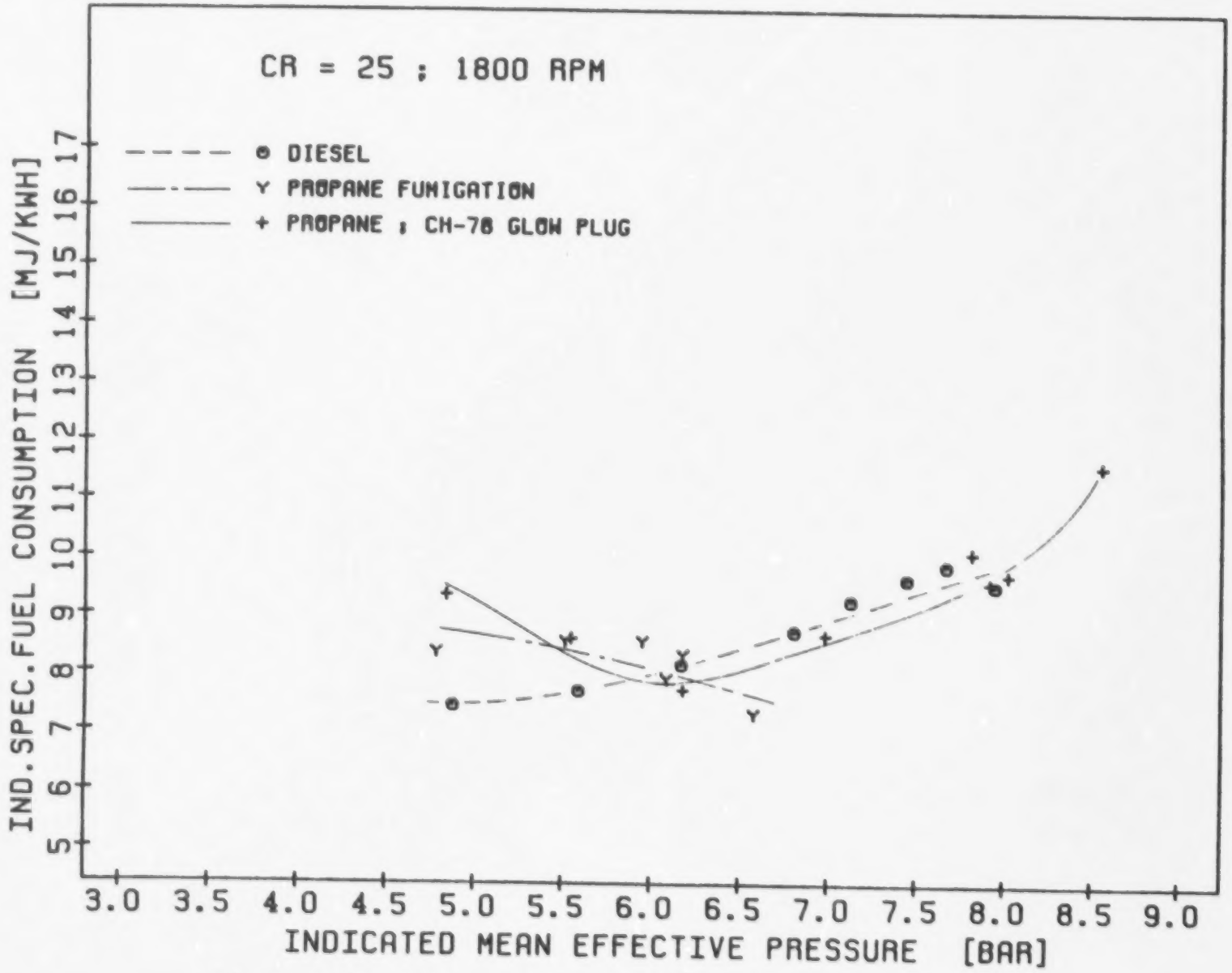

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5 6 7 8 9 10 11 12 13 14 15 16 17
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.27 Indicated specific fuel consumption vs. load as IMEP (CR = 25; 1800 RPM)

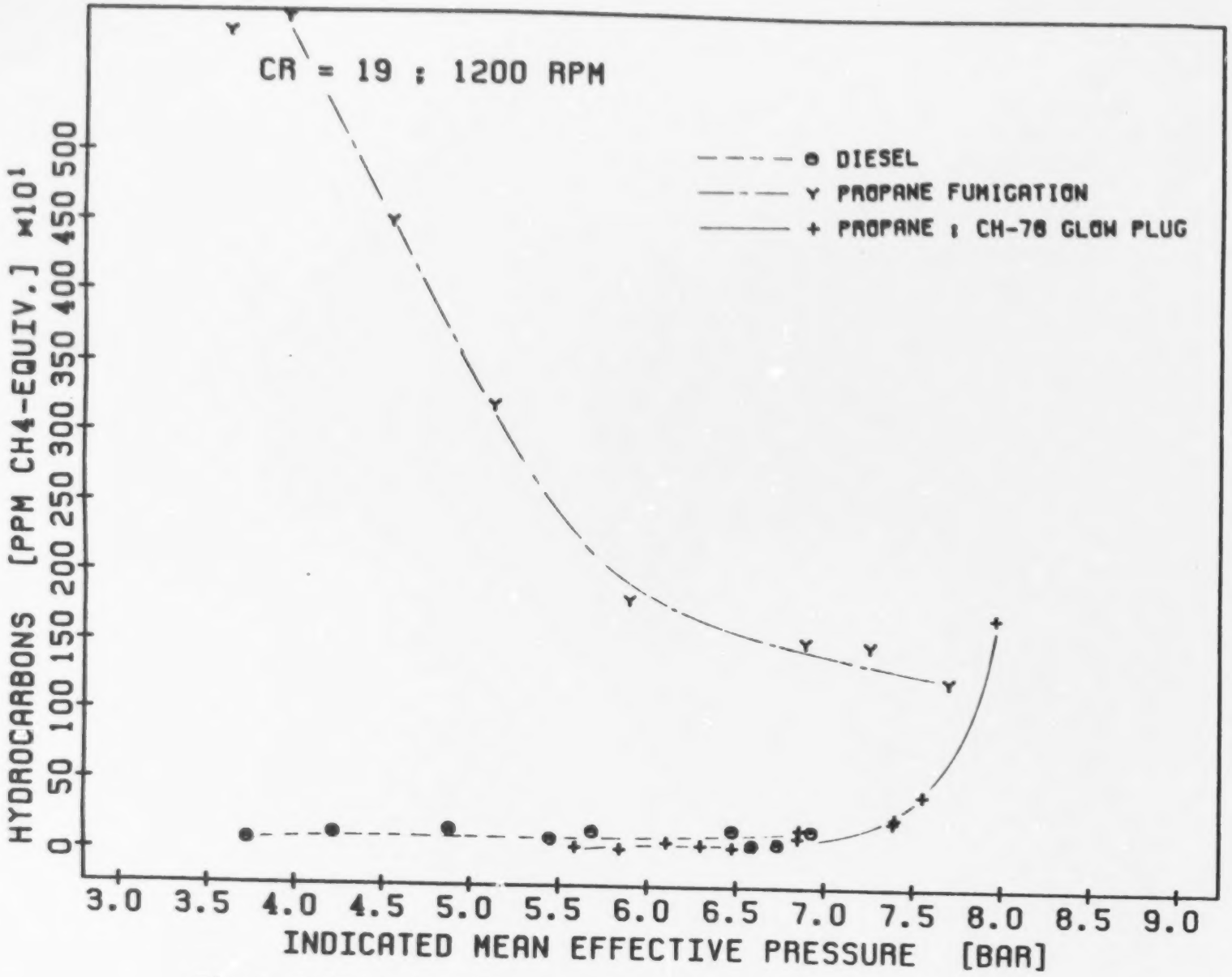

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.28 Hydrocarbons emission vs. load as IMEP (CR = 19; 1200 RPM)

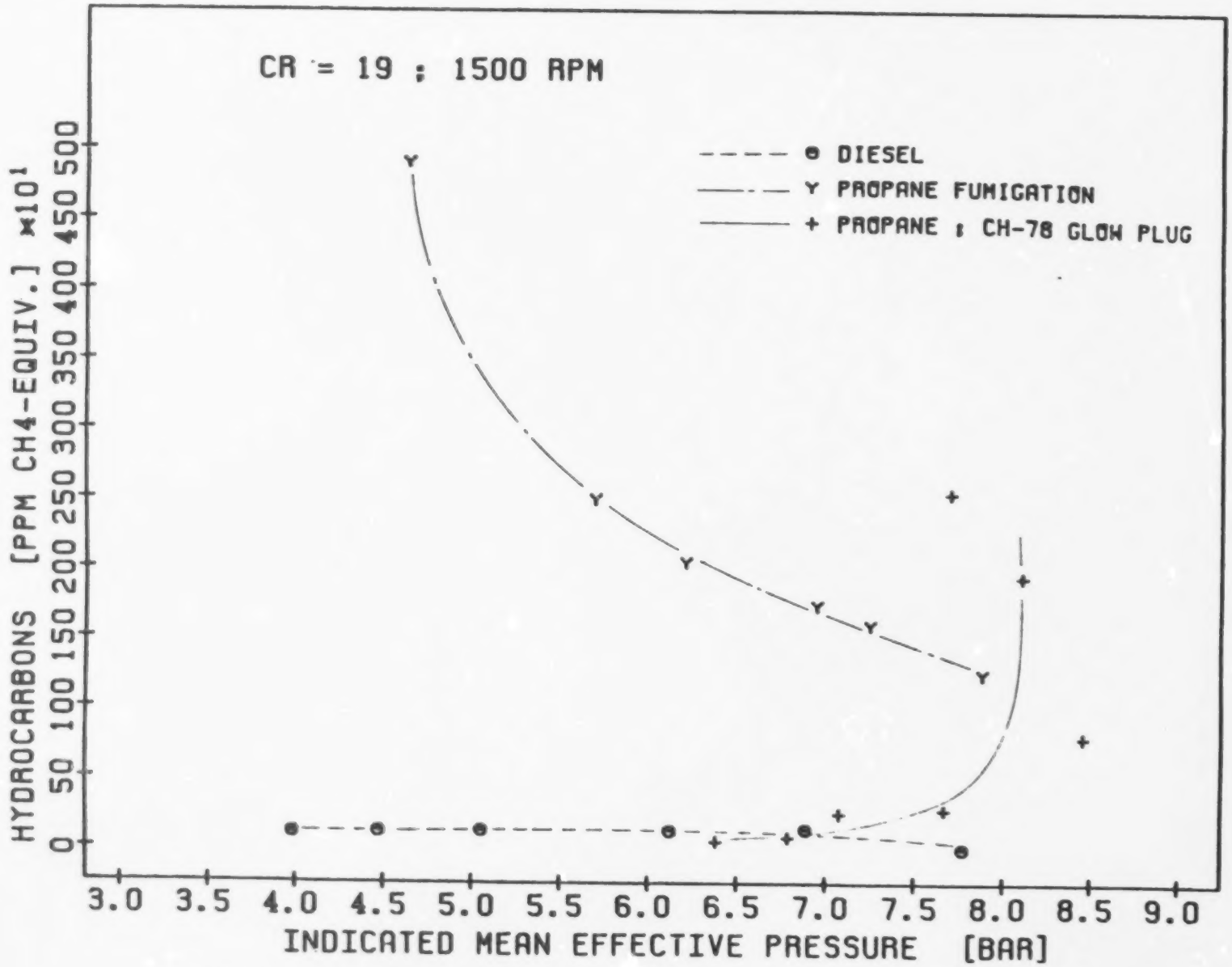

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.29 Hydrocarbons emission vs. load as IMEP (CR = 19; 1500 RPM)

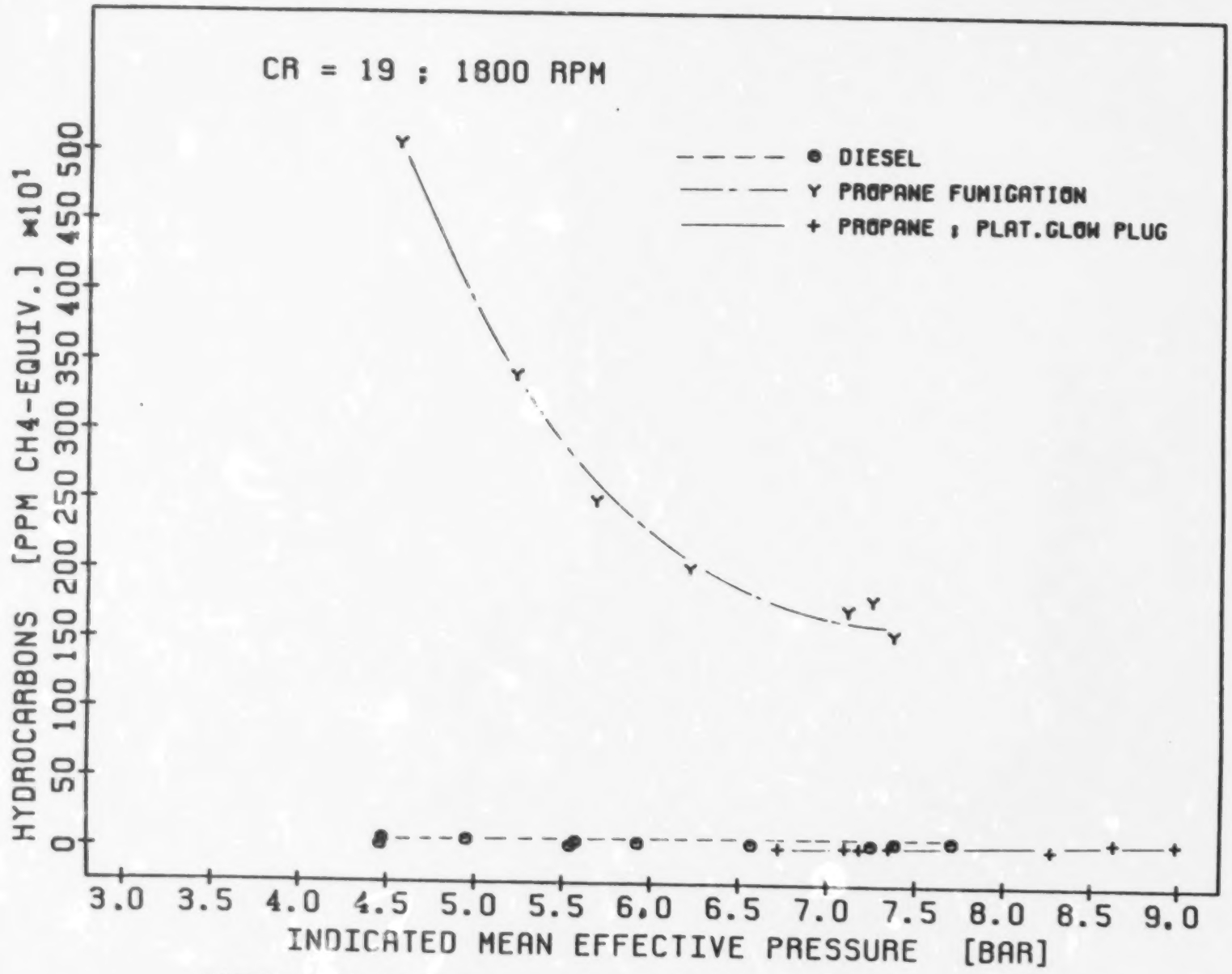

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; PLAT.GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] *10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.30 Hydrocarbons emission vs. load as IMEP (CR = 19; 1800 RPM)

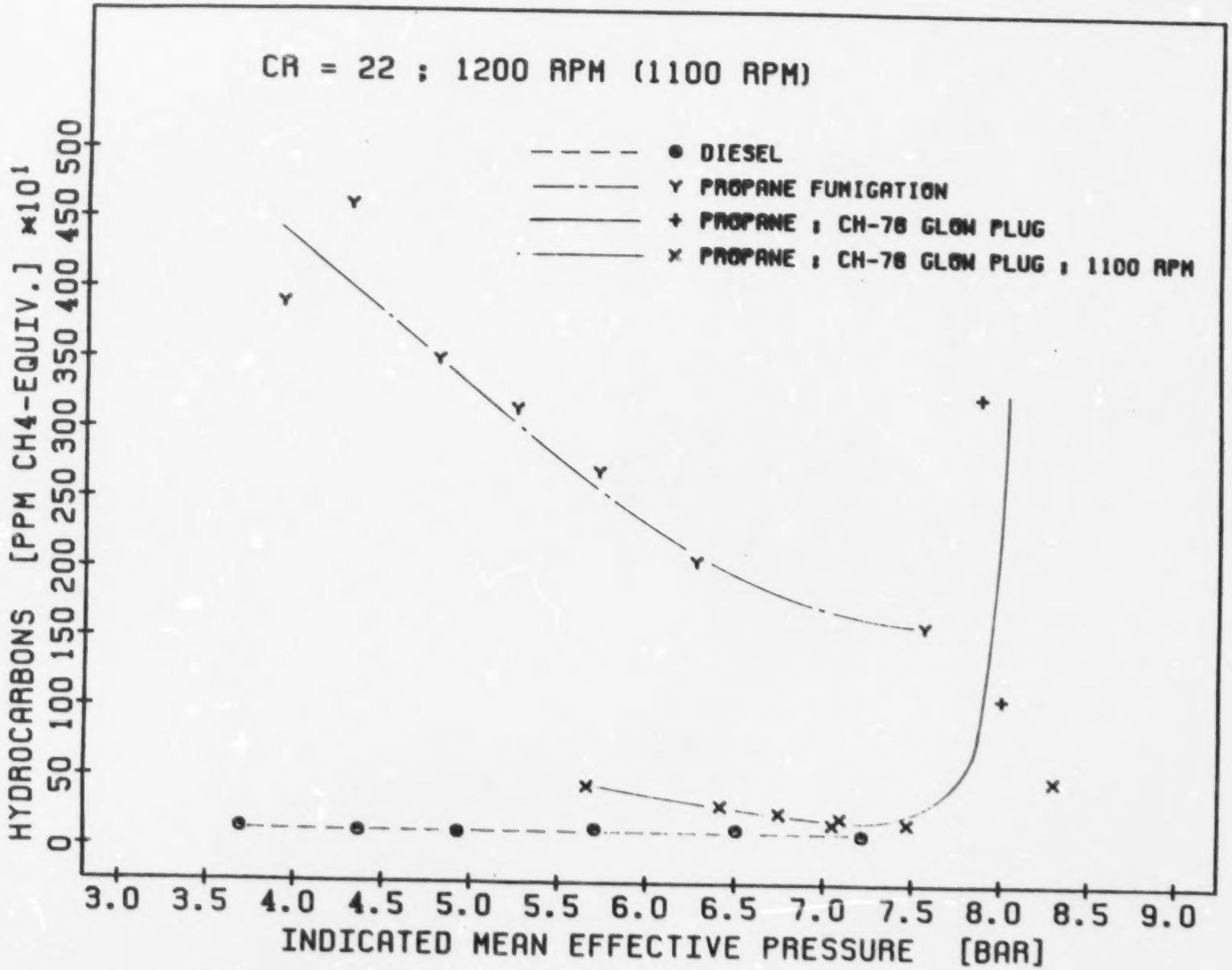

CR = 22 ; 1200 RPM (1100 RPM)
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
PROPANE ; CH-78 GLOW PLUG ; 1100 RPM
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.31 Hydrocarbons emission vs. load as IMEP (CR = 22; 1200 RPM)

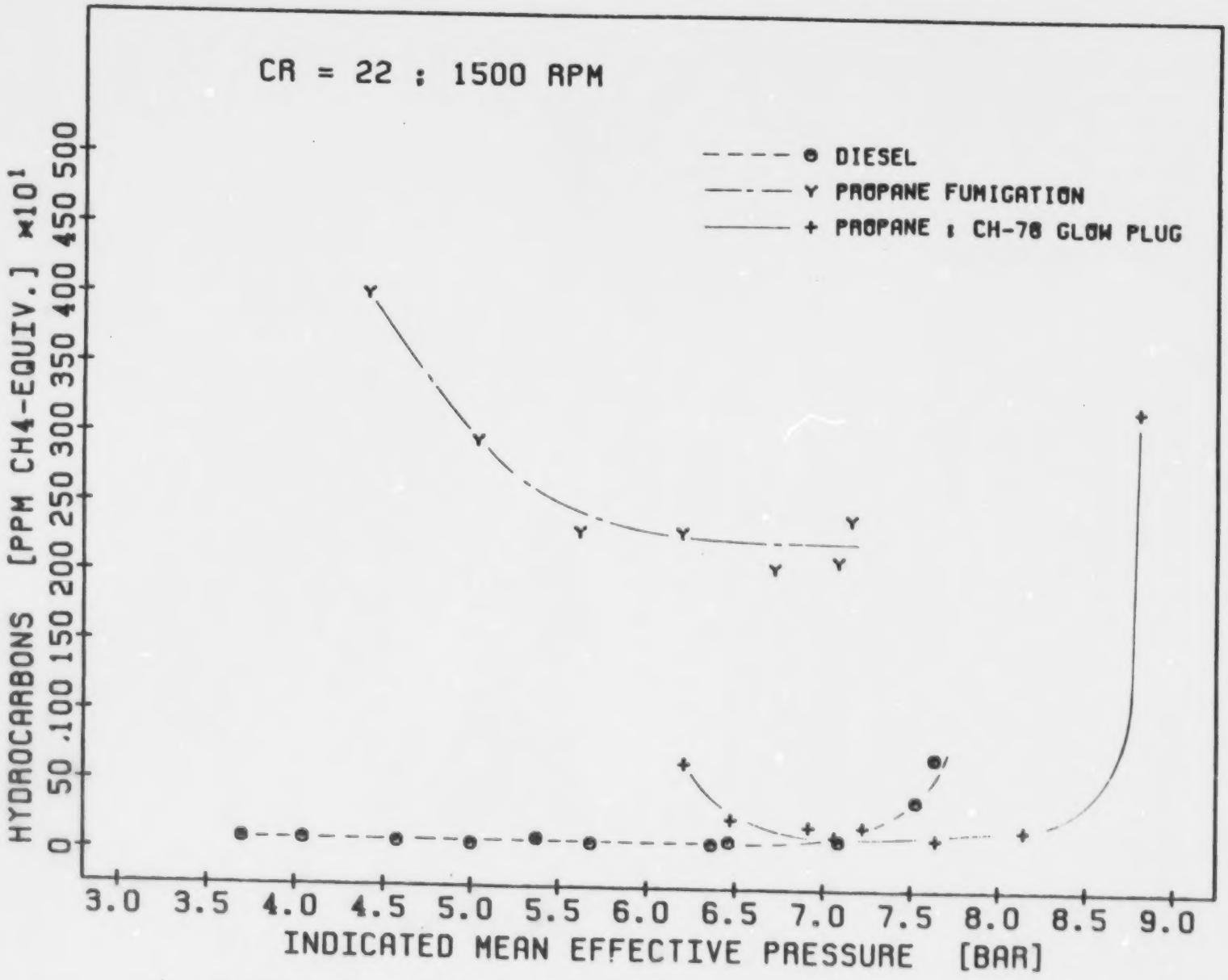

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.32 Hydrocarbons emission vs. load as IMEP (CR = 22; 1500 RPM)

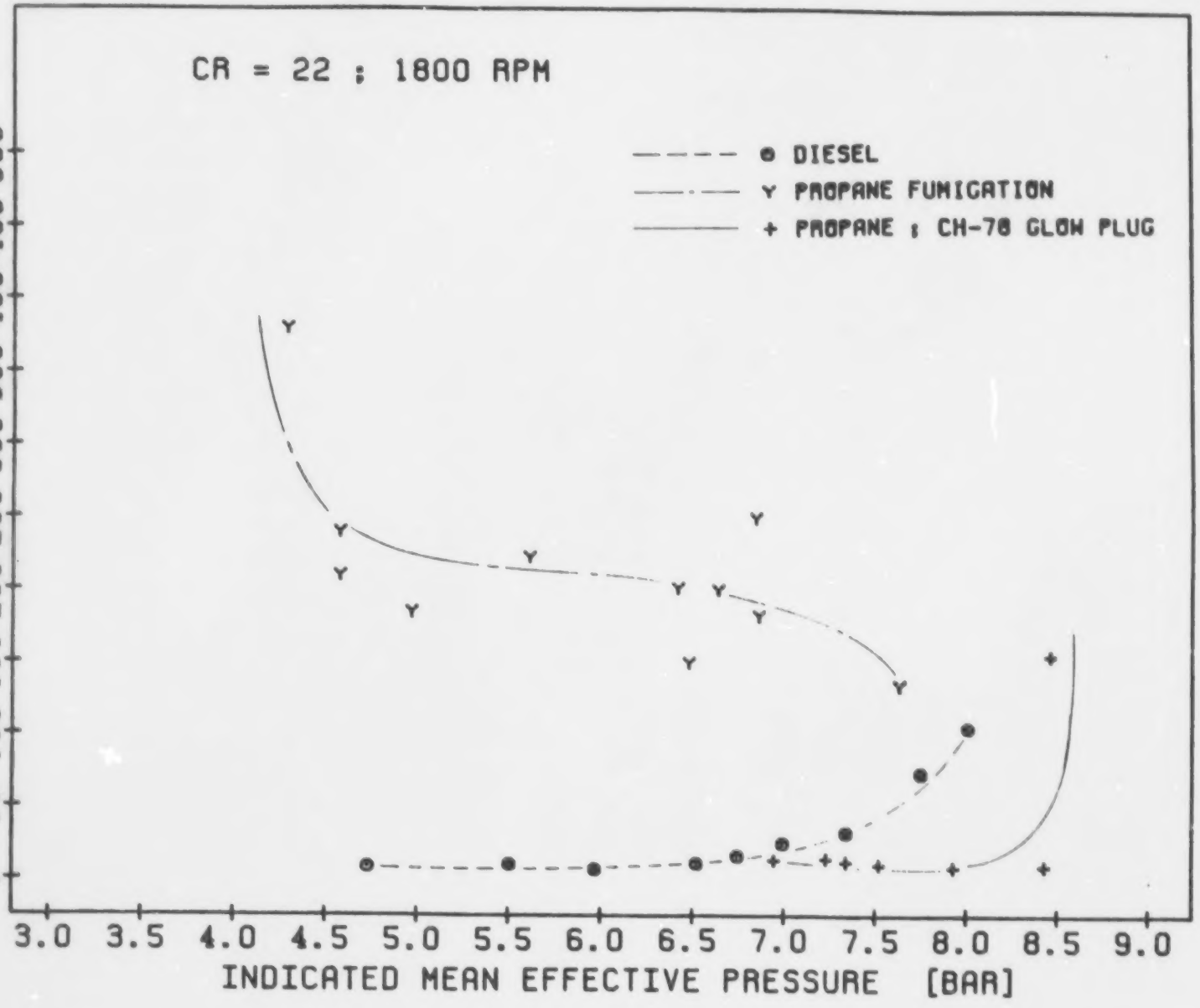

CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] *10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.33 Hydrocarbons emission vs. load as IMEP (CR = 22; 1800 RPM)

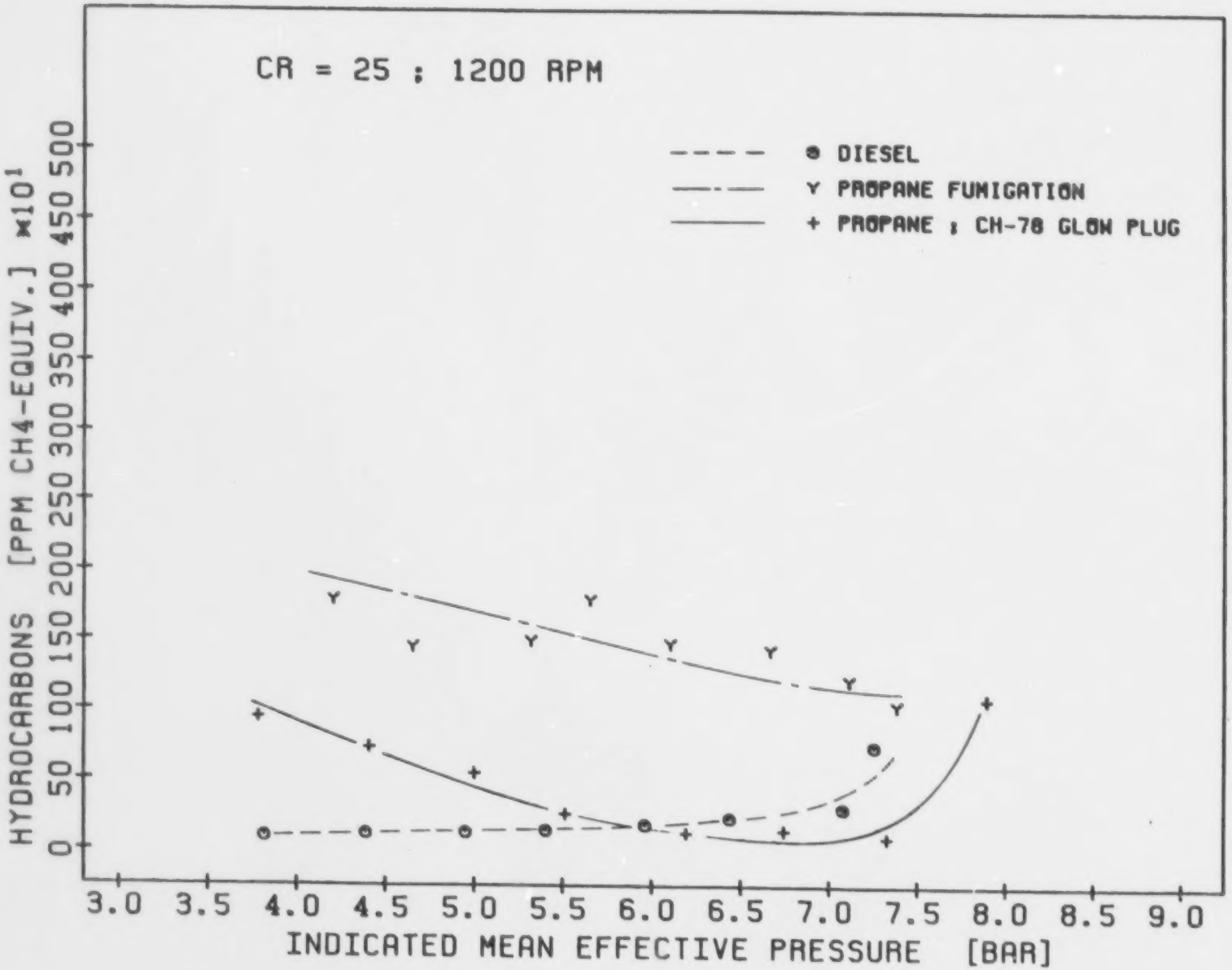

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.34 Hydrocarbons emission vs. load as IMEP (CR = 25; 1200 RPM)

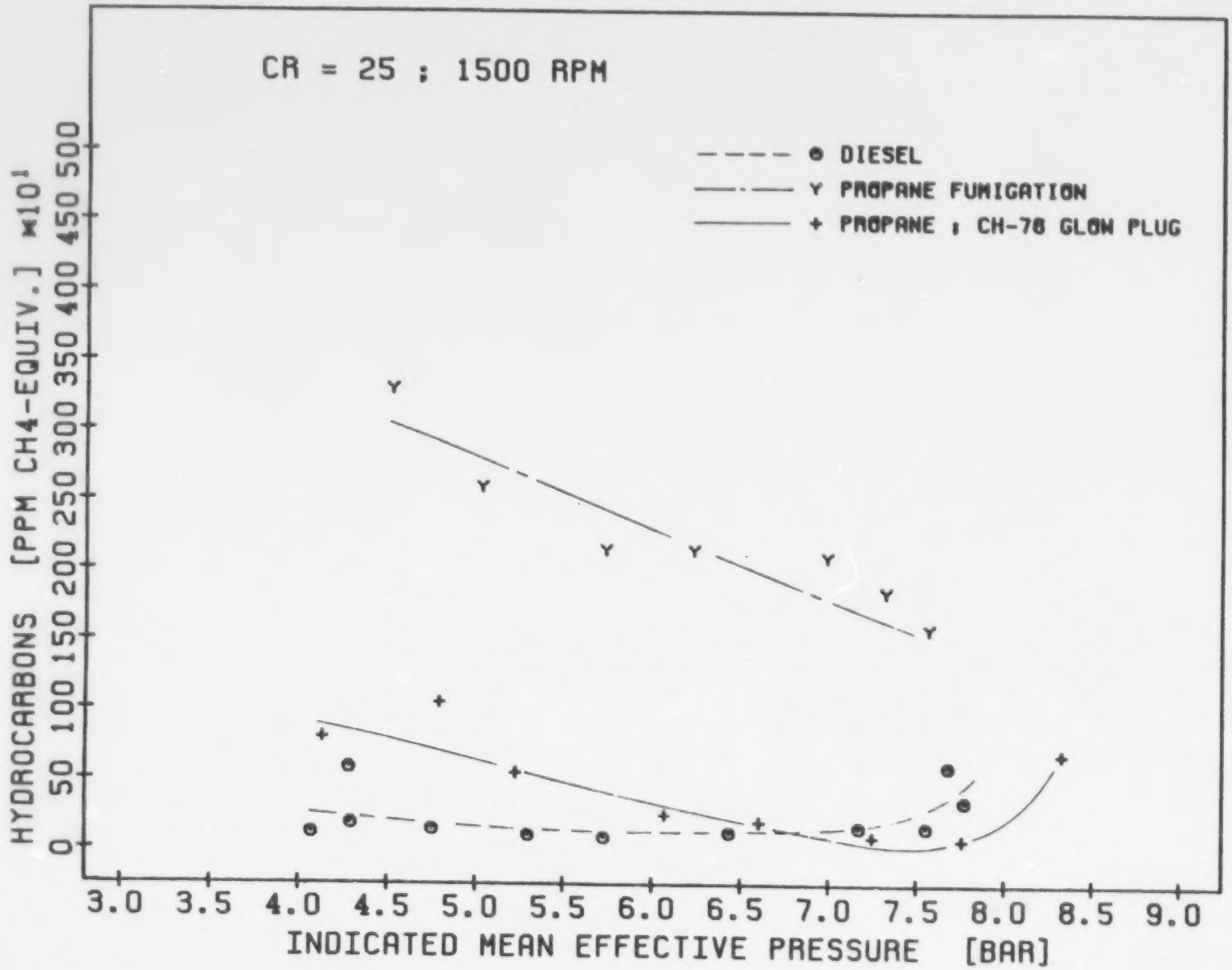

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] *10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.35 Hydrocarbons emission vs. load as IMEP (CR = 25; 1500 RPM)

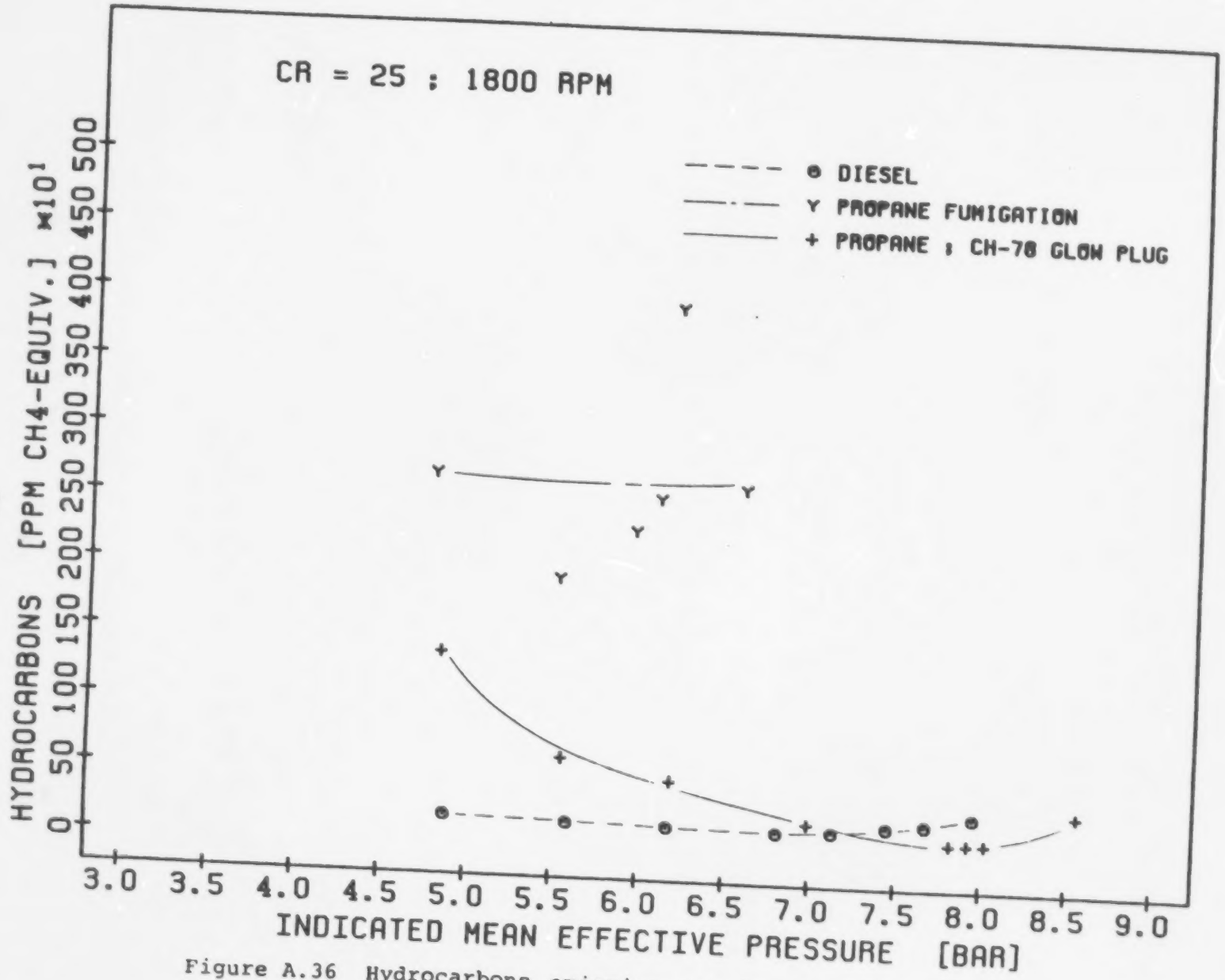

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0 50 100 150 200 250 300 350 400 450 500
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.36 Hydrocarbons emission vs. load as IMEP (CR = 25; 1800 RPM)

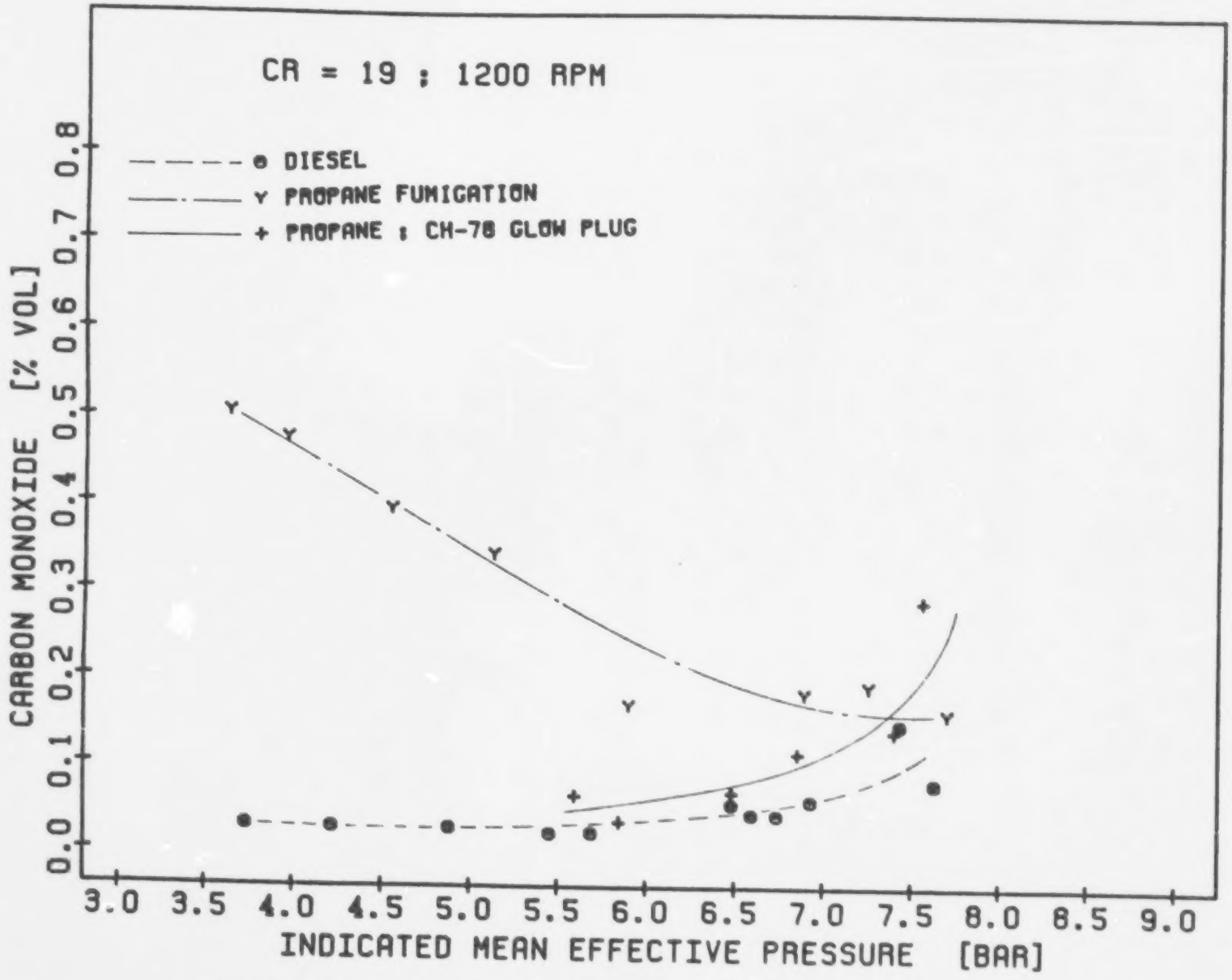

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.37 Carbon monoxide emission vs. load as IMEP (CR = 19; 1200 RPM)

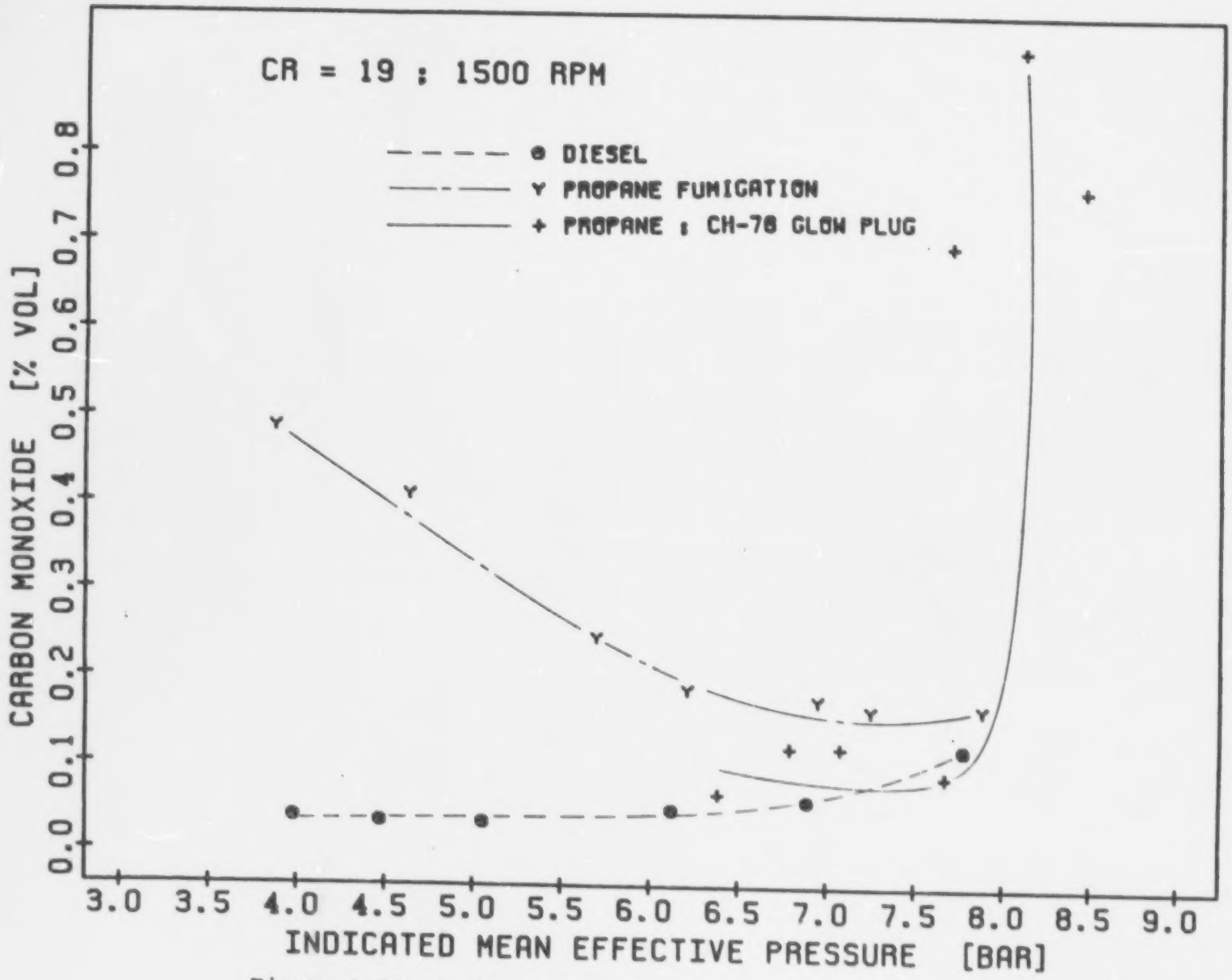

CR = 19 : 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE : CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.38 Carbon monoxide emission vs. load as IMEP (CR = 19; 1500 RPM)

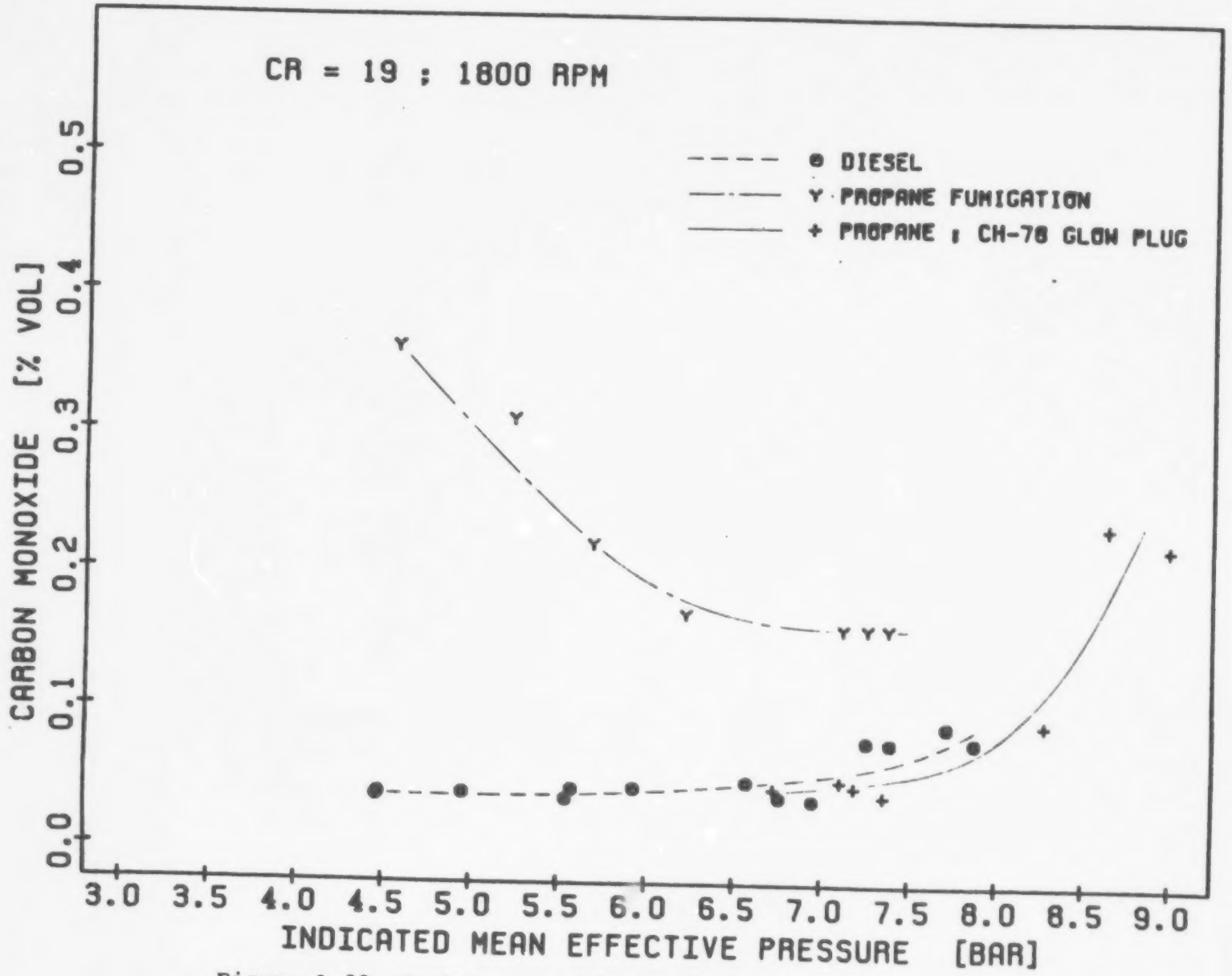


Figure A.39 Carbon monoxide emission vs. load as IMEP (CR = 19; 1800 RPM)

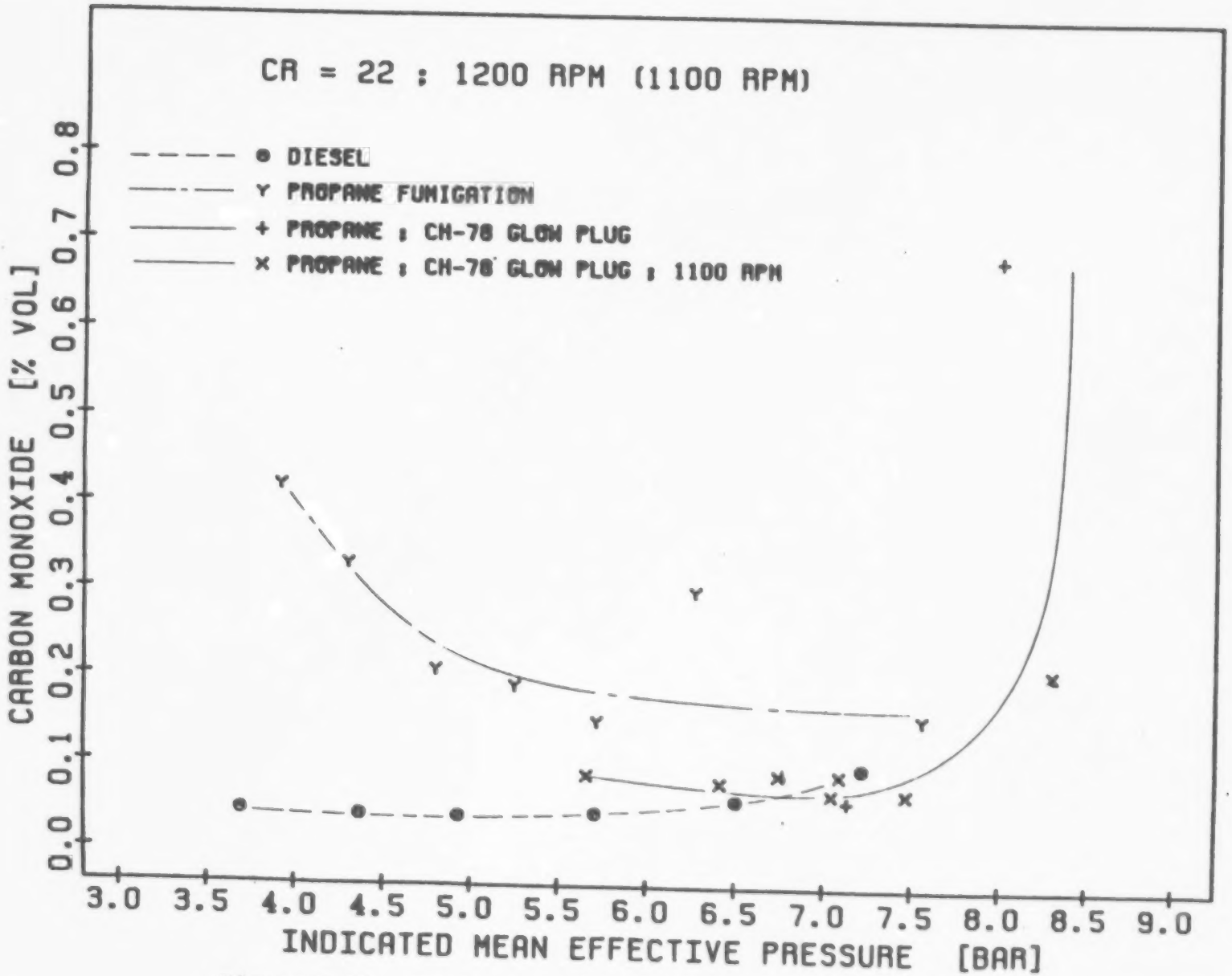

CR = 22 : 1200 RPM (1100 RPM)
DIESEL
PROPANE FUMIGATION
PROPANE : CH-78 GLOW PLUG
PROPANE : CH-78 GLOW PLUG : 1100 RPM
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.40 Carbon monoxide emission vs. load as IMEP (CR = 22; 1200 RPM)

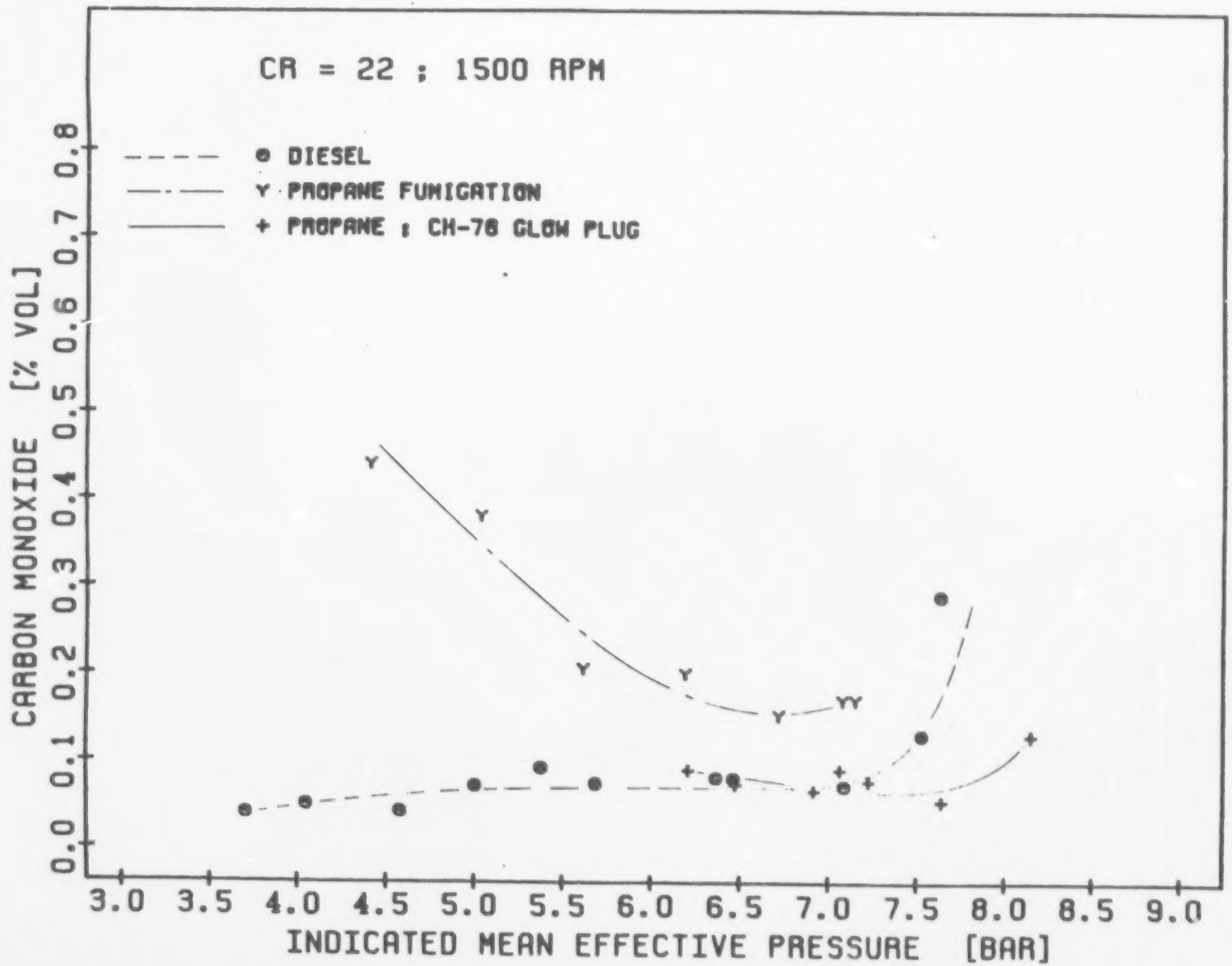

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.41 Carbon monoxide emission vs. load as IMEP (CR = 22; 1500 RPM)

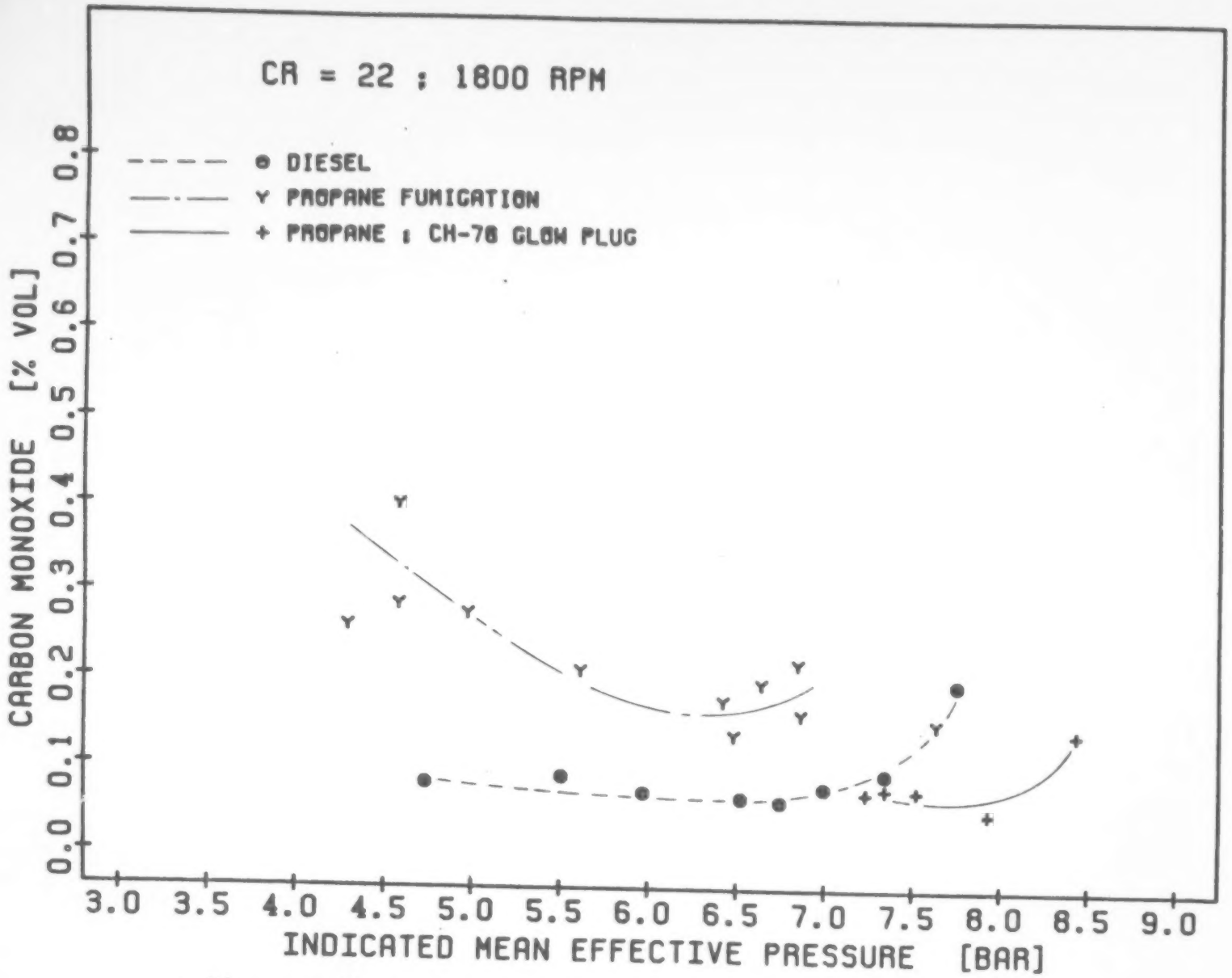

CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.42 Carbon monoxide emission vs. load as IMEP (CR = 22; 1800 RPM)

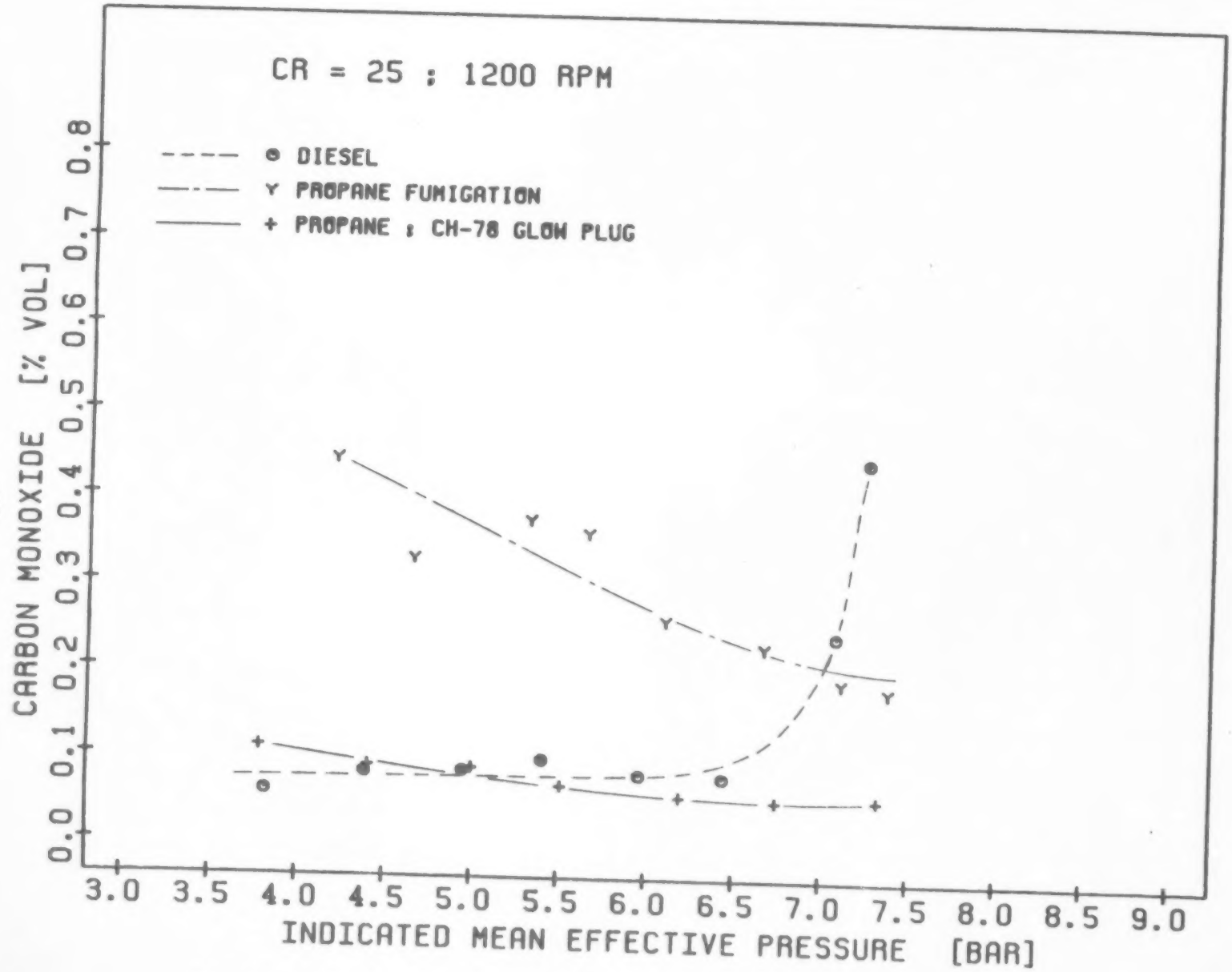

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.43 Carbon monoxide emission vs. load as IMEP (CR = 25; 1200 RPM)

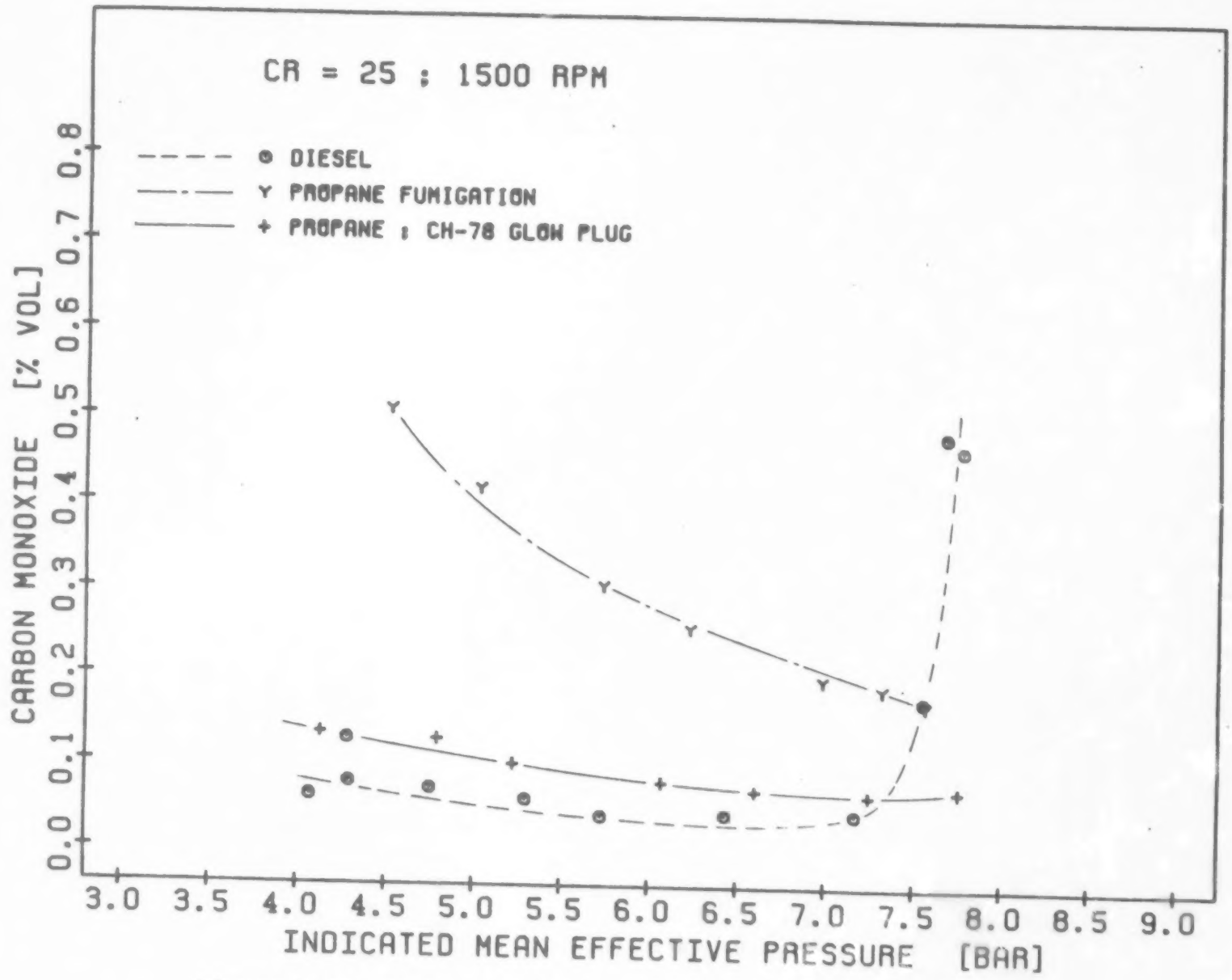

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.44 Carbon monoxide emission vs. load as IMEP (CR = 25; 1500 RPM)

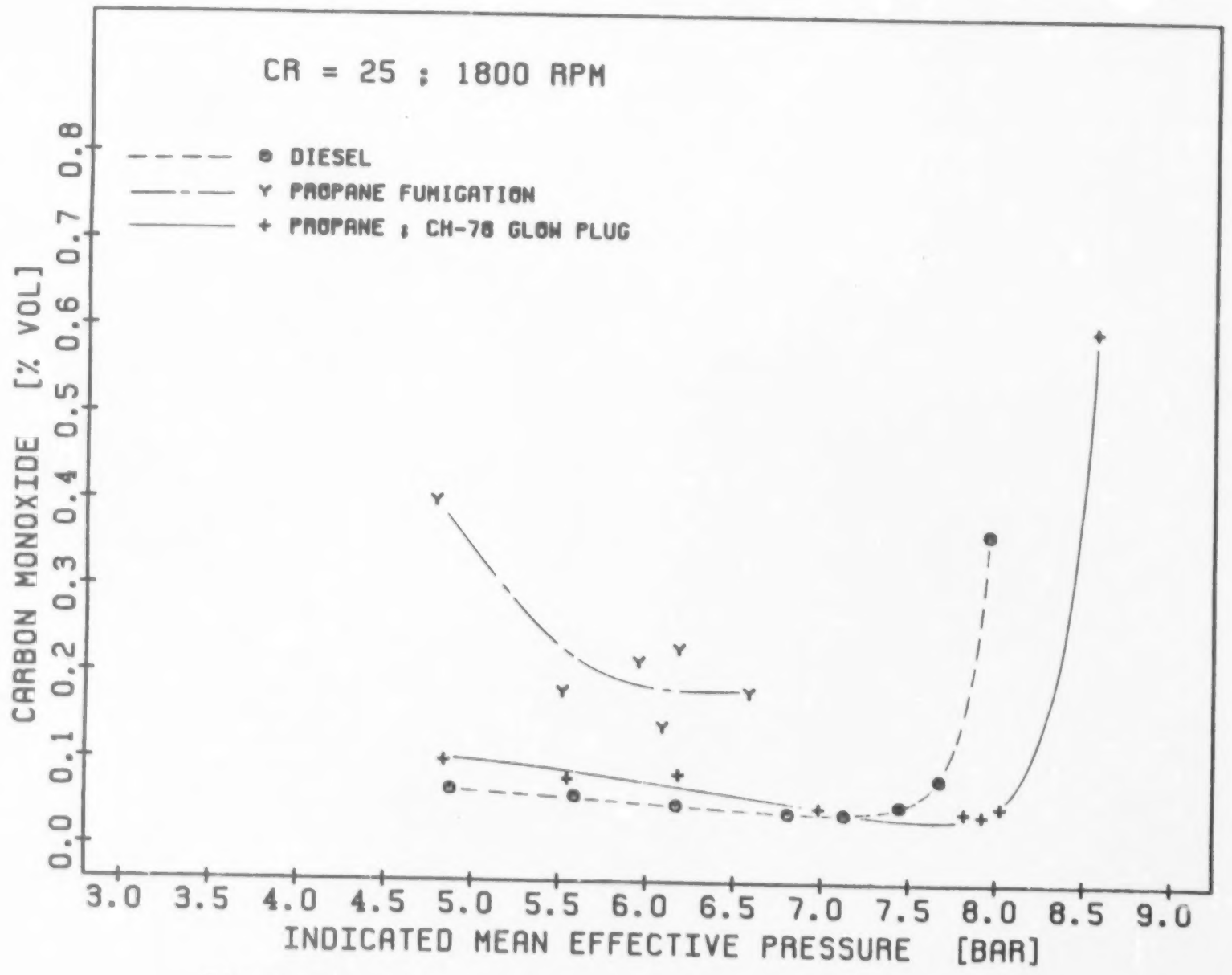

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
PROPANE ; CH-78 GLOW PLUG
CARBON MONOXIDE [% VOL]
0.0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
3.0 3.5 4.0 4.5 5.0 5.5 6.0 6.5 7.0 7.5 8.0 8.5 9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure A.45 Carbon monoxide emission vs. load as IMEP (CR = 25; 1800 RPM)

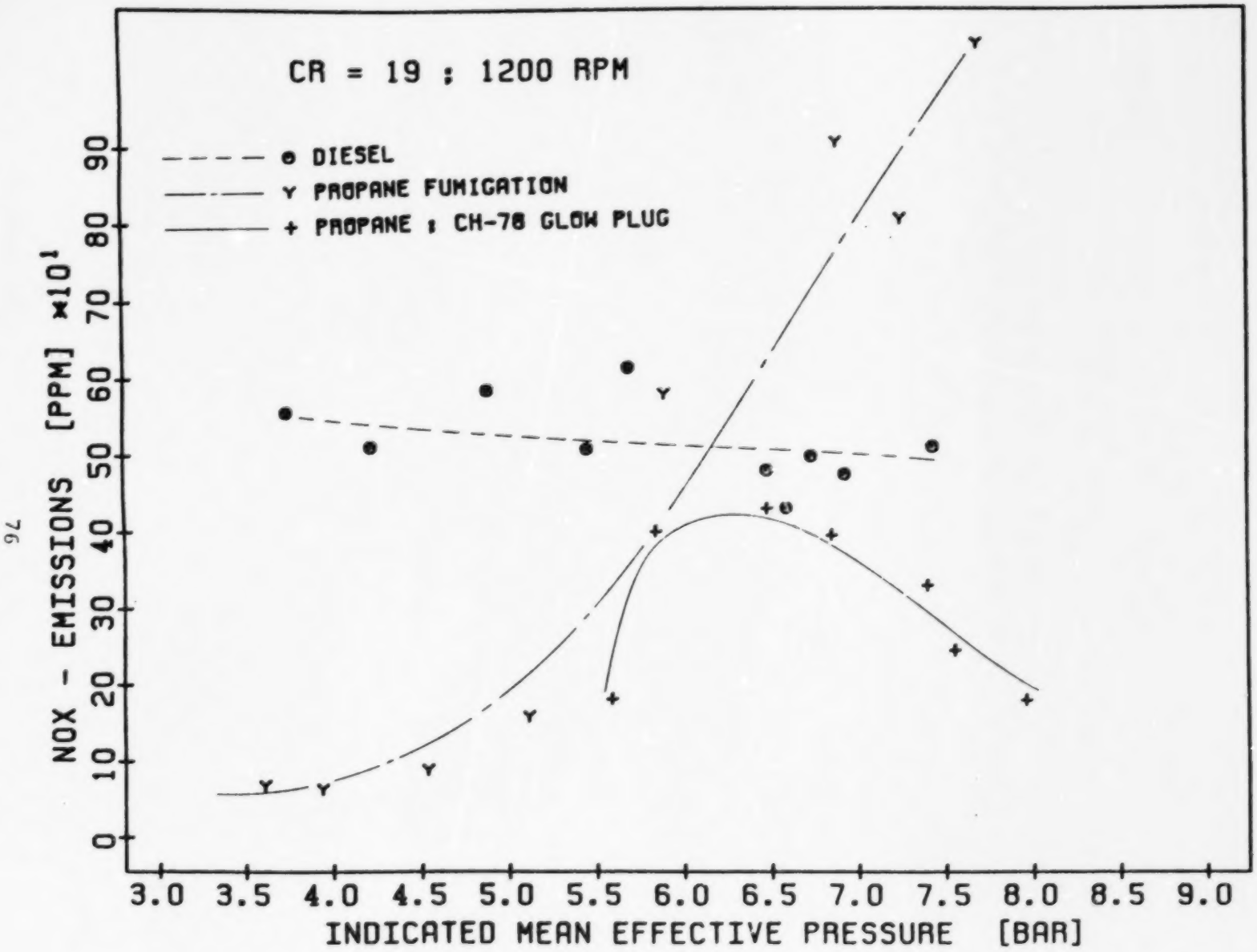


Figure A.46 NO_x emission vs. load as IMEP (CR = 19; 1200 RPM)

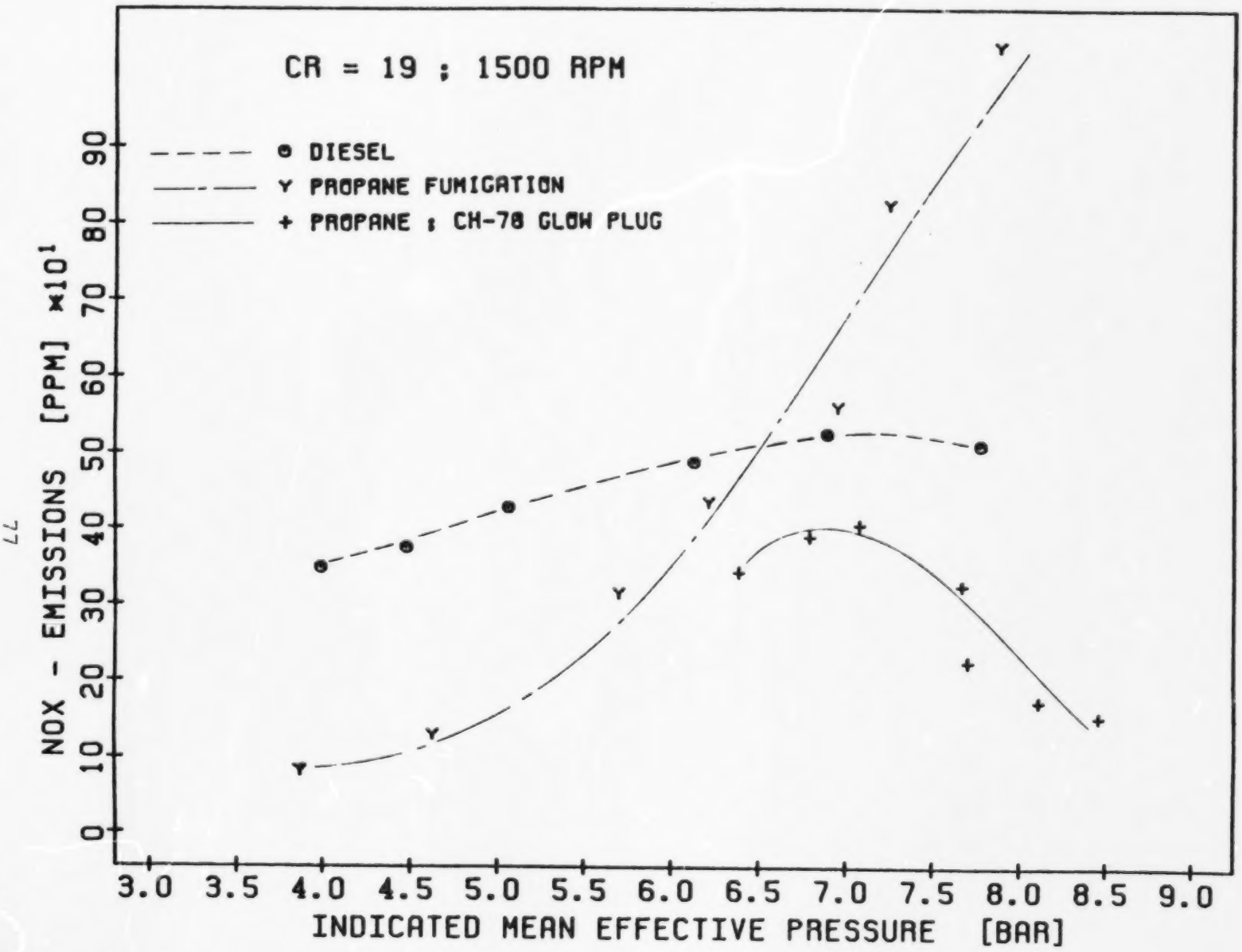


Figure A.47 NO_x emission vs. load as IMEP (CR = 19; 1500 RPM)

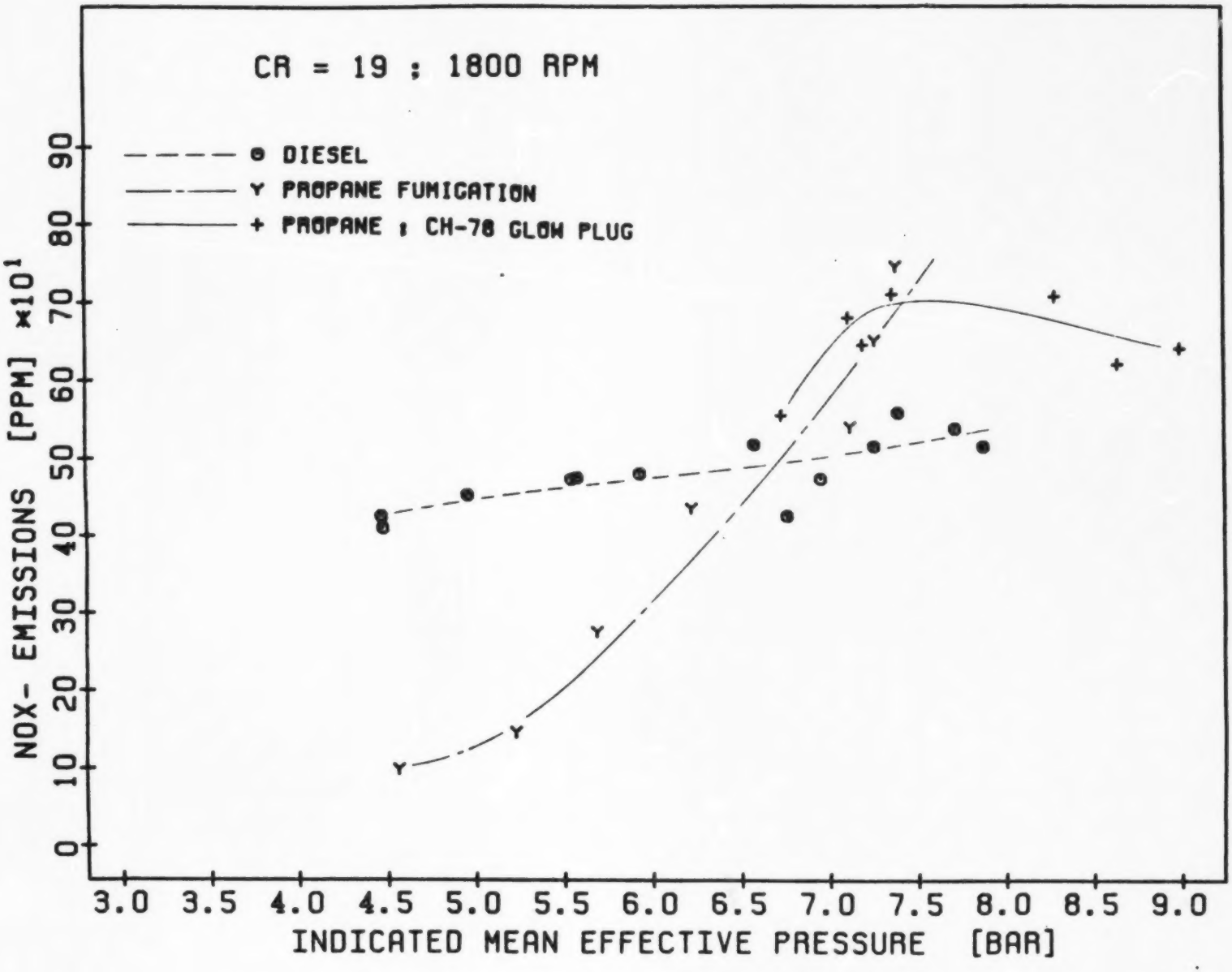


Figure A.48 NO_x emission vs. load as IMEP (CR = 19; 1800 RPM)

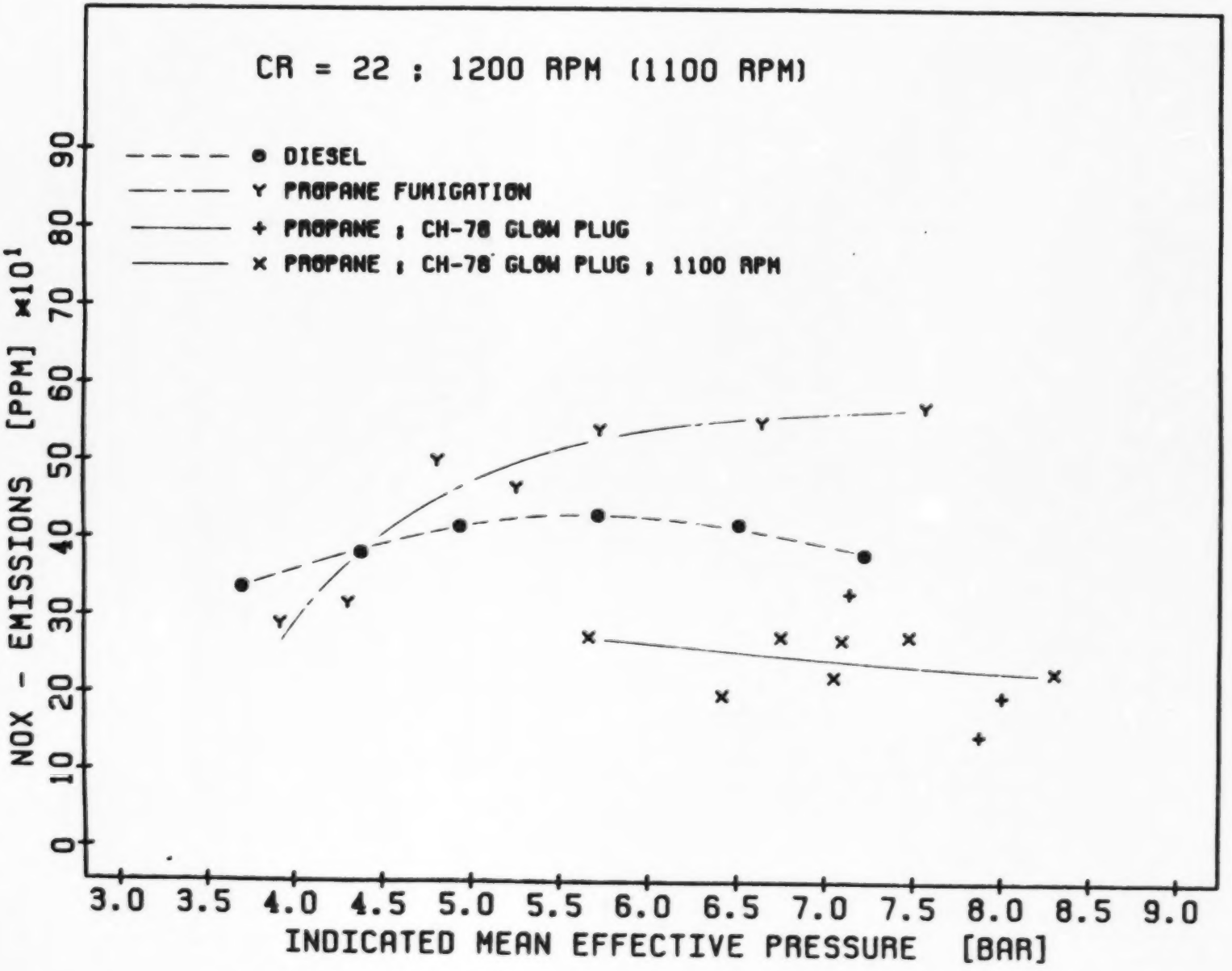


Figure A.49 NO_x emission vs. load as IMEP (CR = 22; 1200 RPM)

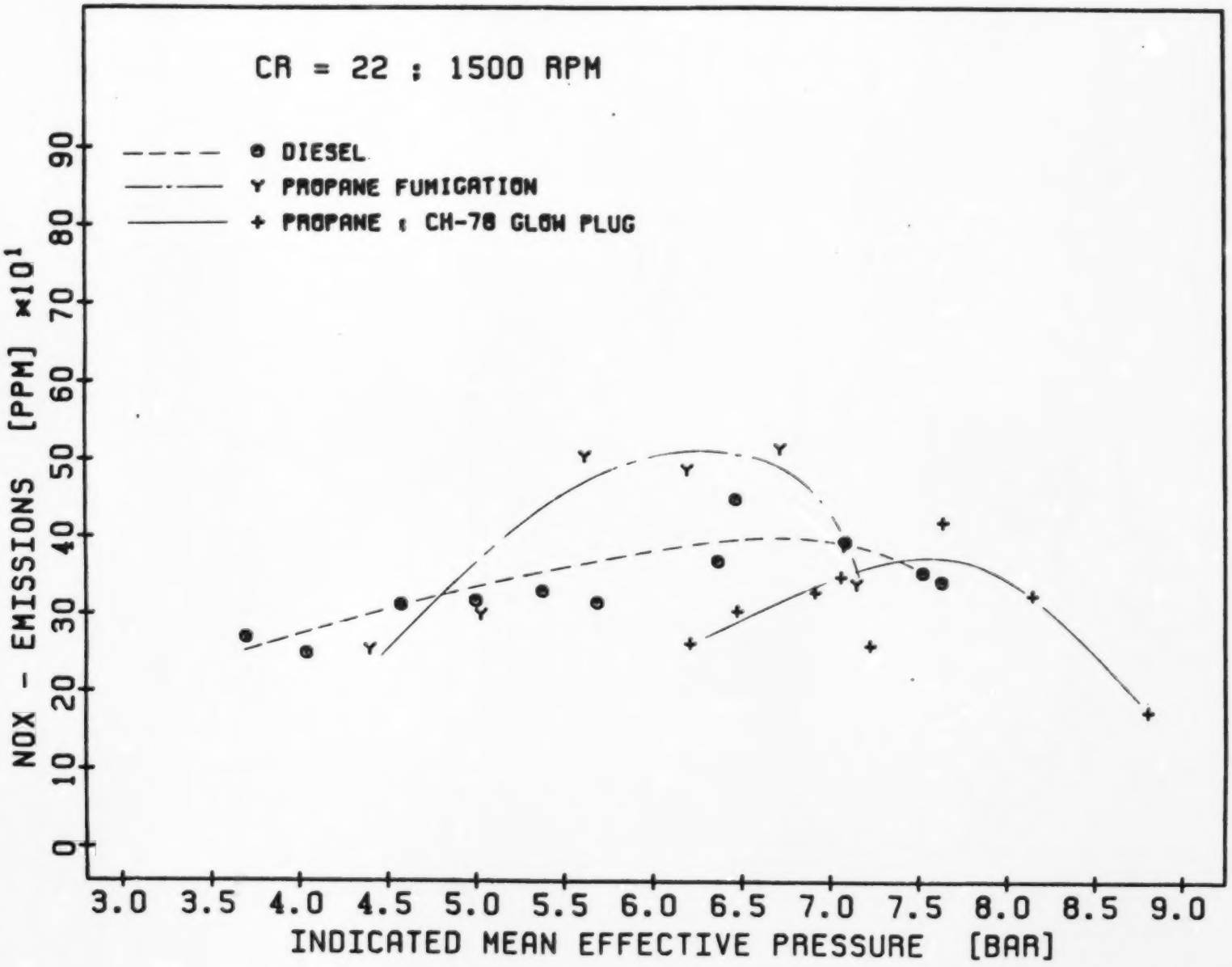


Figure A.50 NO_x emission vs. load as IMEP (CR = 22; 1500 RPM)

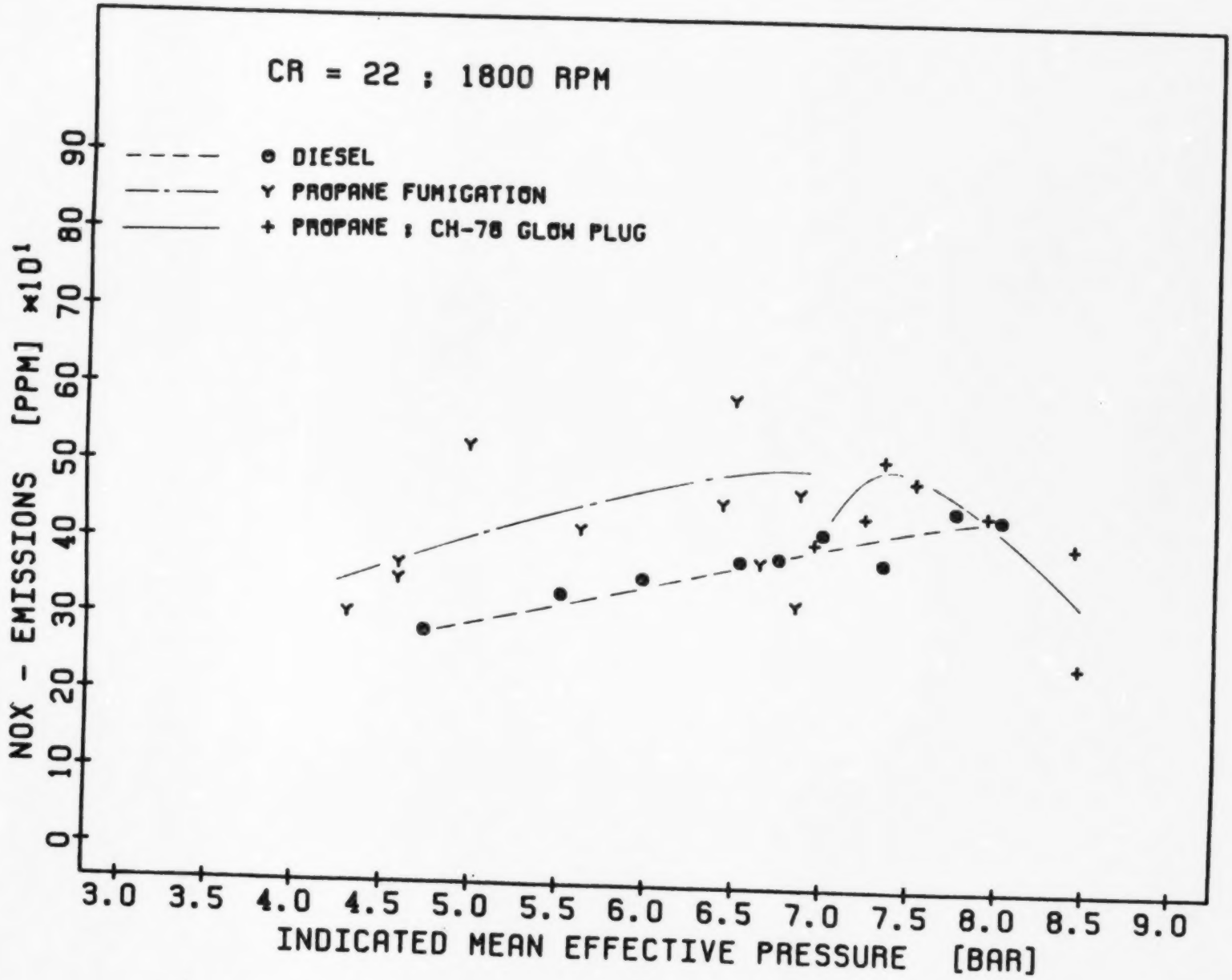


Figure A.51 NO_x emission vs. load as IMEP (CR = 22; 1800 RPM)

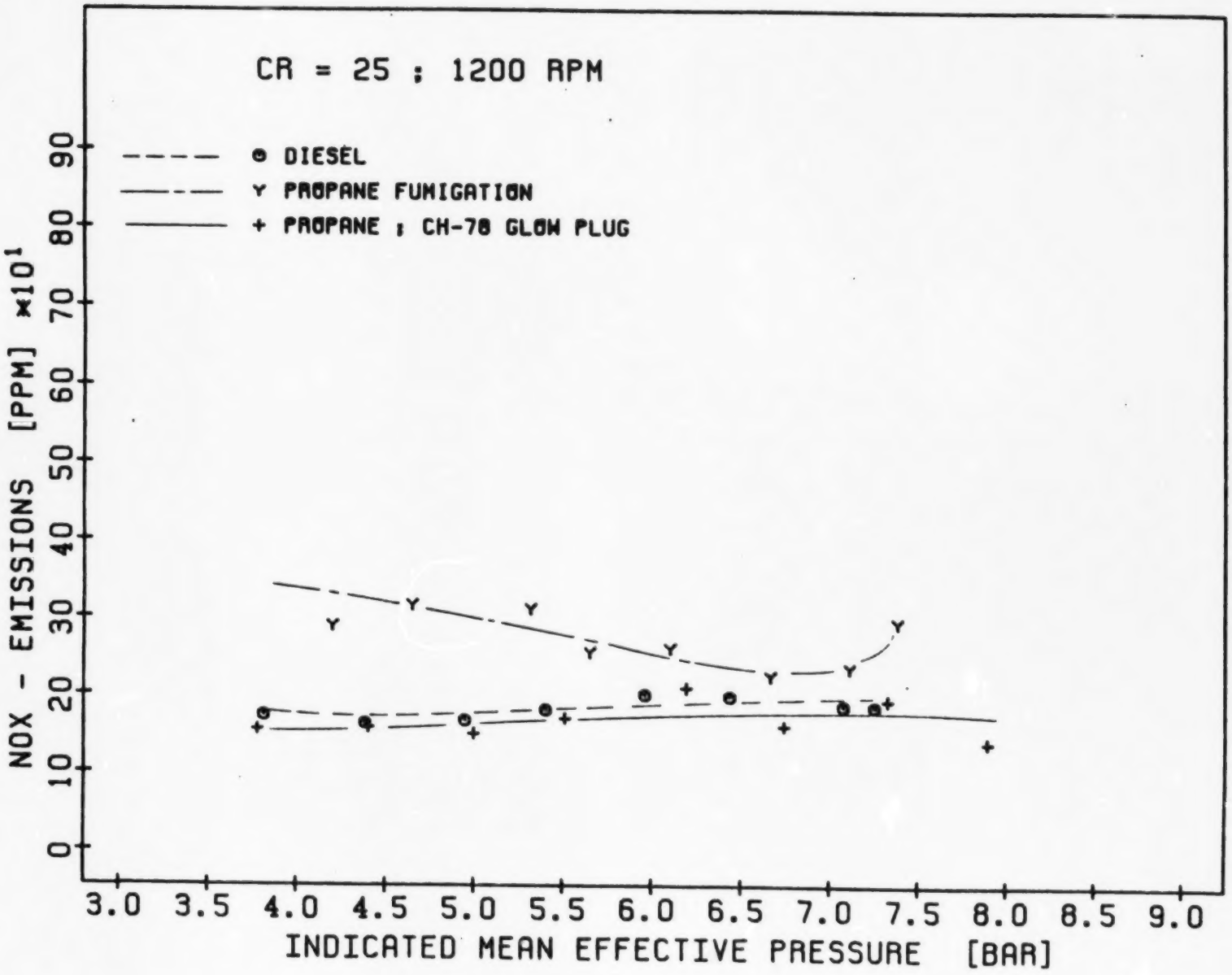


Figure A.52 NO_x emission vs. load as IMEP (CR = 25; 1200 RPM)

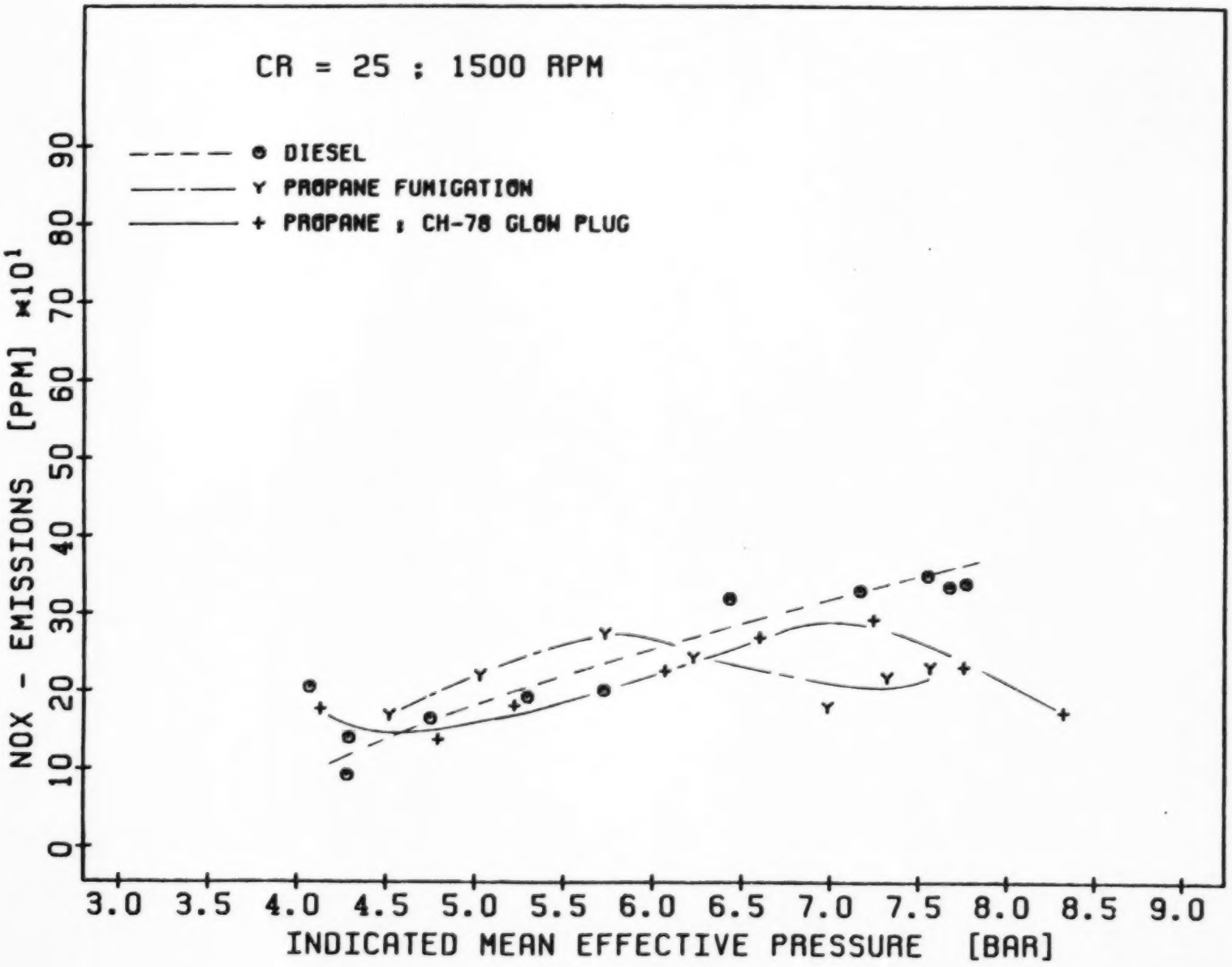


Figure A.53 NO_x emission vs. load as IMEP (CR = 25; 1500 RPM)

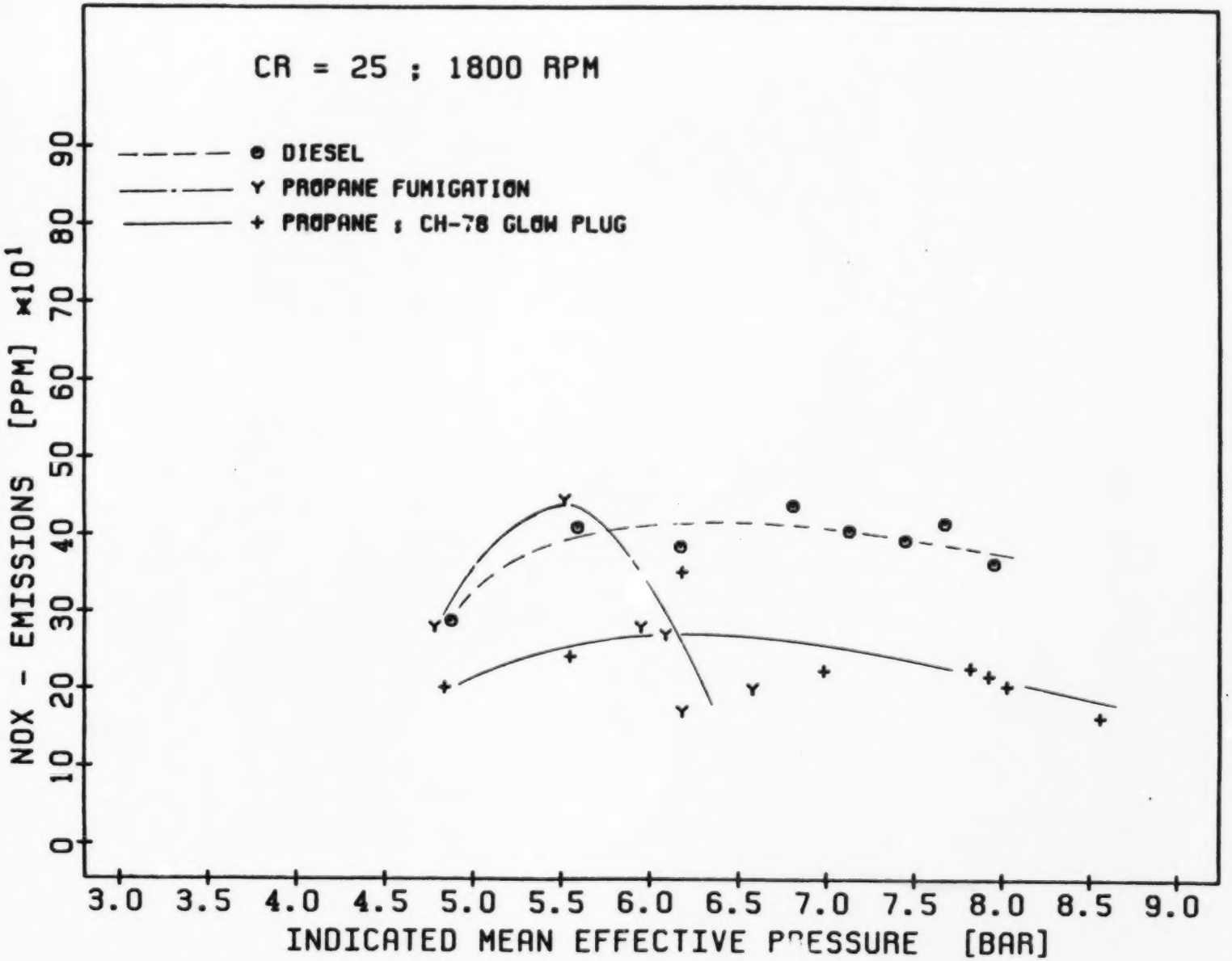


Figure A.54 NO_x emission vs. load as IMEP (CR = 25; 1800 RPM)